THE MAGIC VALVE

BOB NEAL'S EQUALIZER

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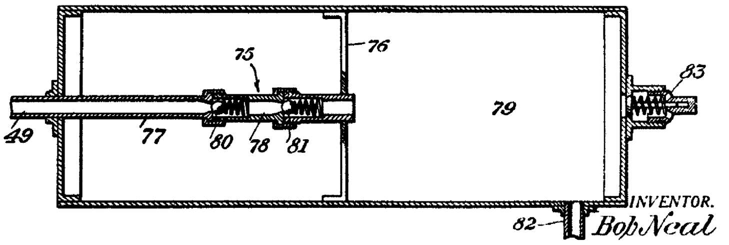


EQUALIZER TANK FROM BOB NEAL'S PATENT, 1936

THE STORM IN A TANK
by Scott Robertson

November 4, 1999

You are looking at this booklet because you might be interested in learning about using compressed air as a medium for making large quantities of solar energy available as engine fuel.

The disadvantages of ordinary solar generating devices--such as bulk, inefficiency, and off-hours--have inspired the search by many for a more widespread and accessible source of solar energy than sunlight, wind, and hydro-power.

It is good that engineers and other solid thinkers want to know about things like *outside* sources of energy as required by the First Law of Thermodynamics, and *continuous* sources of energy as required by the Second Law. The sun is a very good example of an outside, continuous source of energy. Possibly the best example in the solar system. But it is not those elusive solar rays that we should be trying to capture; just go to the dump where wasted light beams pile up, and guess what.

The First Law has decreed that there is no destruction of energy, so the waste product of light energy--and a lot of other energy media--is mostly heat. Ambient heat is the heat in the air around us. The sun invests 460° F. worth of heat in our atmosphere just to get it to 0° F. You know how much energy exists in a small box containing air heated to 500° if you've ever left a melted cheese sandwich in the oven 30 seconds too long.

After 99% of the light beams from the sun have failed to strike a solar panel, bounced off the earth and dissipated their energy by helping to heat up the four quadrillion tons of Earth's atmosphere, there sits this sea of "totally" dissipated energy proving its inability to do work by blowing around as weather. Next time a hurricane blows down your house, don't worry, it just wasn't possible. Ambient heat is "totally" dissipated.

Wrong.

THE AUTHOR DRIVING AN AIR POWERED GO-KART THAT HE DESIGNED AND BUILT IN TEN HOURS, JULY 1988

All we need is a little weather inside a tank full of compressed air. The atmosphere of this planet carries easily available clean solar energy, and the compressed air engine uses heat energy as-is without the need to convert it to something else. The self-fueling air engine is actually a heat pump that uses compressed air as its working medium instead of freon. This makes it possible to replace bulky, expensive heat exchangers with direct mixing of atmosphere into the compressed air already in the tank. Most conventional heat pumps can provide three times more heat energy to their application than the amount of electrical energy used to operate them. With a storm in a full air tank, we expect to be able to match that or better. There is more energy in a tank full of compressed air than is needed to start an air engine and keep the tank full for a few seconds. All we need is a way to replace the solar energy leaving the tank with more solar energy on its way into the tank so that after a few seconds, the tank pressure has not gone down, and the process of continual replenishment can go on indefinitely.

This cannot be done with conventional compression cycles, because it is considered wise and practical in these early days of the engineering science to waste heat as quickly as possible, so that all of the compressor's effort is used to waste heat to the earth's atmosphere. What is needed is a device or process that can provide for the continual replenishment of an air tank by *injection*--not compression--of large quantities of atmosphere into the tank. The compression needed is done by mixing of the atmosphere in the tank with the compressed air that was already in there. The result is a whole new batch of compressed air every few seconds, compliments of the sun.

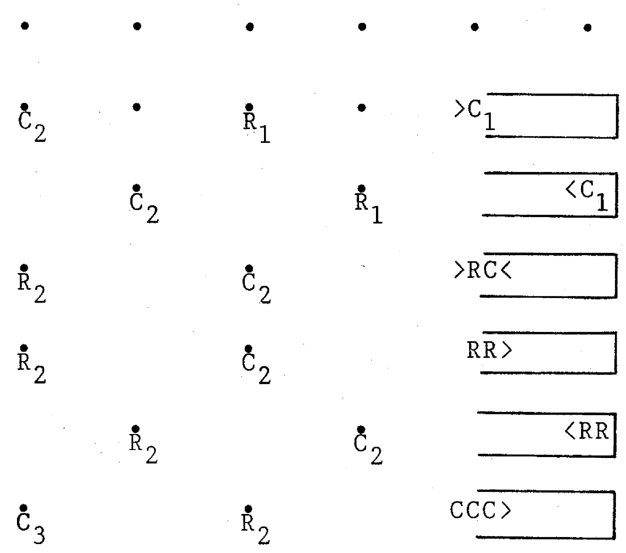
James Clerk Maxwell, who was the discoverer of the electromagnetic spectrum and (according to Albert Einstein) the father of modern physics, first outlined a plan for running air engines on free ambient heat, in his textbook *Theory of Heat* in 1870. The key to his concept was to set up two zones of unequal pressures in an air tank by some mysterious uninvented process, and then run an air engine on the difference. Like an organized storm. Put a storm in a tank and you don't need a compressor.

It's just another way to say that the energy that pushes pistons is heat, not pressure as we assume. The conventional compressor, unbeknownst to most engineers, does not invest its work in pressurizing an inert substance to produce this "pressure" which we imagine to be capable of doing work. As many textbooks of the earlier "hands-on" era of pneumatics will tell you, ALL COMPRESSION WORK IS LOST AS HEAT and therefore the work done by an air engine is derived from the heat the air carried before it was compressed. The act of compressing air into a tank is not an investment, but an attempt to put something where it doesn't want to go, an attempt to achieve the impossible, so it's not surprising that all the work done is wasted. Only by luck is it possible to smash air together, so some air ends up trapped in the tank after all this fuss is over and all the compression work has become friction and floated away into the atmosphere as more wasted energy.

So had to go and refer to his mysterious uninvented process as a "little being" in a tank who sorts the air molecules into two separate zones of high and low energy content. The little being became the heart of the debate amongst the physics departments, and it became the task of ignorant slobs like me to realize that there is enough energy in a full tank of air to not only run the car but to also force/sneak a jetstream of sun-warmed atmosphere in through the back door while the physics clique is busy arguing about exactly why it can't be done: *the little demon is blind...the little demon is computer illiterate...*so many reasons the little demon is incapable of sorting molecules. But what about the two zones of unequal pressures? What about the kind of hardware and processes that routinely create zones of unequal pressure? All forgotten in the race to publish the most unintelligible paper.

Let me be the first to pat myself on the back for learning the answer to Maxwell's riddle, which is: ***Any process that converts pressure into velocity and back into pressure with minimum energy loss can be adapted to get low pressure air into a pressurized air tank against relatively low resistance.*** Candidates for Maxwell's Demon, that is, examples of processes that convert pressure into velocity, are: the Bernoulli Effect, the Venturi Effect, the Coanda Effect, and sound waves. I have thousands of pages of documentation showing that these phenomena are routinely used for a large

Fig. 1) resonance in 1/4- or 3/4-wave closed-end tubes

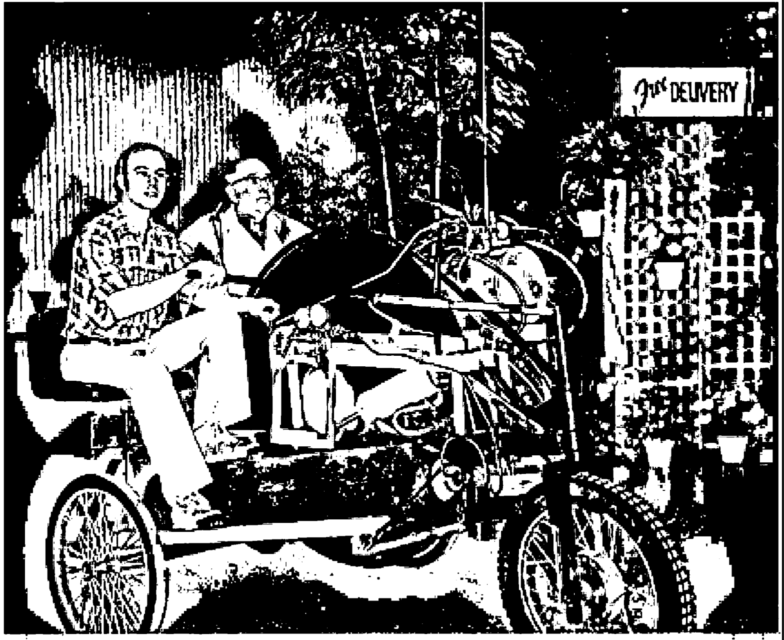


SKETCH BY A MECHANICAL ENGINEER SHOWING THE ACOUSTIC PHENOMENA OF WAVE REFLECTION, WHICH COULD MAXIMIZE THE EFFECTIVENESS OF THE EQUALIZER IF IT WERE EMPLOYED IN DESIGN.

variety of exotic processes closely related to the process of creating an organized storm in a tank. The point is that, when we stop thinking of compressed air as the by-product of a compressor, and start thinking of it as a carrier of solar energy, then it suddenly behooves us to focus on any unusually affordable method of raising air's pressure to a usable level so its internal energy (heat content) can be used to run an air engine for free. We no longer must necessarily draw an automatic mental connection between elevated pressure and the forceful squashing together of molecules into a confined space. We can do with air what we've been doing with steam for 150 years: we can put low pressure air into a high pressure tank. Once it's in there, its pressure, or ability to expand, becomes elevated by mixing with what was already in there, and its energy, or ability to do work, is provided by the heat in the atmosphere, which is replenished constantly by the sun.

I have one last complaint about the assumptions we make about compressed air. When I say "heat pushes pistons," I also imply that pressure does not. Pressure is a gauge of compressed air's ability to expand and push pistons, but pressure is only a sub-component of the air's energy. Energy is the ability to do work, whereas pressure is only a component of force, and force is only a component of work, making pressure two steps removed from being qualified to do any work. Similarly, when pressure is converted into velocity, the velocity is only a component of momentum, and momentum is only a component of the kinetic energy that actually gets the fresh air into the full tank without a struggle. It is heat that provides the energy needed to push pistons, and it's only because of the sun that the average tank of air is able to do anything at all.

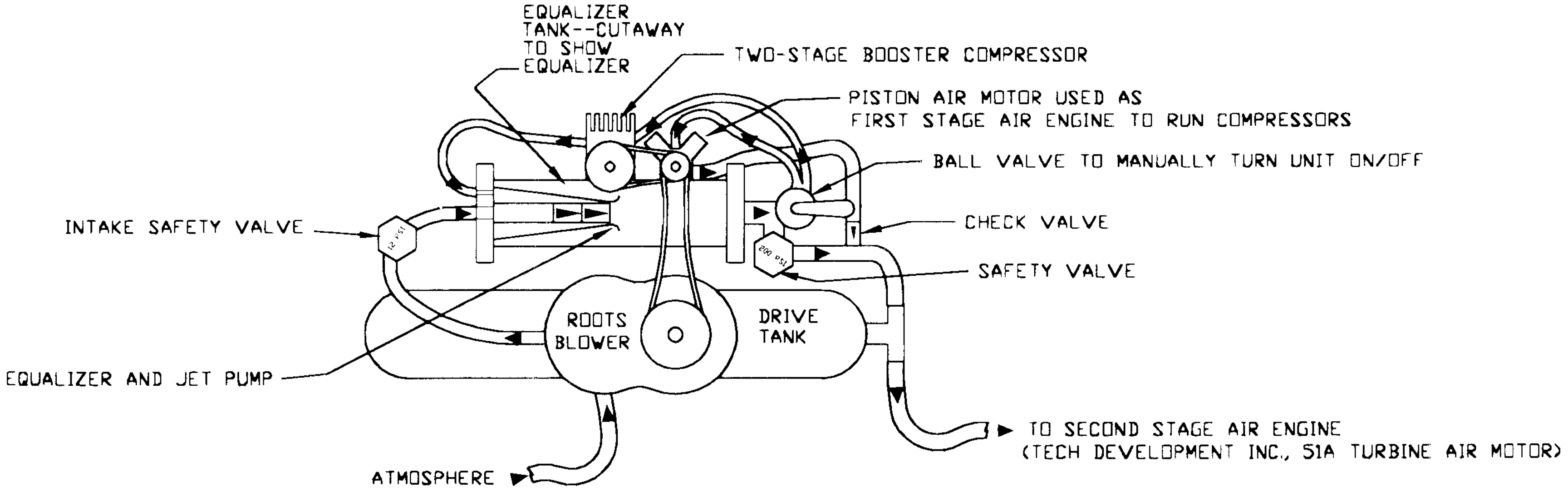
Understanding these few basic ideas about how compressed air works is essential to being able to use air to its best advantage, and its best advantage is that **COMPRESSED AIR IS SOLAR ENERGY.** All we need is a magic valve in an air tank that temporarily converts pressure into velocity, a natural process known as the Bernoulli Effect, to get low pressure air into a high pressure tank, without wasting a lot of stored air to get it done. Is that so hard?



WICHITA, KANSAS, 1985: THE AUTHOR DRIVING TERRY MILLER AROUND IN CIRCLES IN TERRY'S PROTOTYPE AIR CAR, WHICH HE BUILT IN 1979 FOR $1500 AND DEMONSTRATED FROM COAST TO COAST. TERRY'S AIR CARS HAVE BEEN WRITTEN UP IN SEVERAL NATIONAL MAGAZINES AND TECHNICAL JOURNALS.

MODIFIED NEAL COMPRESSION UNIT

BASED ON BOB NEAL'S U.S. PATENT NO. 2,030,759


EQUALIZER TANK--CUTAWAY TO SHOW EQUALIZER
TWO-STAGE BOOSTER COMPRESSOR
PISTON AIR MOTOR USED AS FIRST STAGE AIR ENGINE TO RUN COMPRESSORS
BALL VALVE TO MANUALLY TURN UNIT ON/OFF
INTAKE SAFETY VALVE
12 PSI
CHECK VALVE
200 PSI
SAFETY VALVE
ROOTS BLOWER
DRIVE TANK
EQUALIZER AND JET PUMP
TO SECOND STAGE AIR ENGINE (TECH DEVELOPMENT INC., 51A TURBINE AIR MOTOR)
ATMOSPHERE

NOT SHOWN: PRESSURE
GAUGES OR SENSORS,
FLOW METERS, AIR
CLEANERS & LUBRICATORS

NOT TO SCALE

SELF-FUELING PNEUMATIC POWER PLANT FOR TESTING

DESIGNED BY SCOTT ROBERTSON
AIR CAR RESEARCHER SINCE 1979

PNEUMATIC OPTIONS P.O. Box 2342
Grass Valley, CA 95945

MODIFIED NEAL COMPRESSION UNIT

based on Bob Neal's U.S. Patent No. 2,030,759

KEY COMPONENTS OF TEST CONFIGURATION

MANUALLY OPERATED ON/OFF BALL VALVE
TANK FULL OF COMPRESSED AIR
INTAKE PORT AT ONE END OF TANK
DISCHARGE PORT AT THE OTHER END OF TANK
SMALL TWO-STAGE BOOSTER COMPRESSOR RUN BY SMALL PISTON AIR MOTOR THAT CAN OPERATE AGAINST DOWNSTREAM BACK PRESSURE
EQUALIZER, A SERIES OF TWO CHECK VALVES IN TANK INTAKE LINE JUST UPSTREAM FROM WHERE JET PUMP CAUSES INTAKE AIR TO ENTER AND MIX WITH BOOSTER AIR
TWO SAFETY VALVES, ONE TO VENT TANK INTAKE LINE IF OVER 10 PSIG, AND ONE TO VENT EQUALIZER TANK TO DRIVE TANK IF PRESSURE EXCEEDS 200 PSIG
JET PUMP, THE HEART OF THE DEVICE
PRESSURE GAUGES, ONE IN COMMUNICATION CHAMBER OF EQUALIZER, ONE IN BLOWER DISCHARGE PIPE AND ONE IN MAIN TANK AREA
POSITIVE DISPLACEMENT BLOWER OR LARGE VOLUME PISTON COMPRESSOR. AMBIENT PRESSURE IS BOOSTED BY HIGH-VOLUME-LOW-PRESSURE (HVLP) COMPRESSOR TO ASSIST ENTRY INTO SUCTION PORT OF JET PUMP. ROOTS BLOWER CAN BE USED IF JET PUMP SUCTION ACHIEVES 10 PSIG OR LESS; OTHERWISE A PISTON COMPRESSOR RUN AT A RELATIVELY HIGH SPEED WILL PROVIDE POSITIVE PRESSURE TO THE INTAKE CHECK VALVE OF THE EQUALIZER. IF ROOTS BLOWER IS USED, A SAFETY VALVE IS INSTALLED IN ITS DISCHARGE LINE TO VENT TO ATMOSPHERE IF LINE PRESSURE EXCEEDS 12 PSIG.

MAIN EVENTS OF SYSTEM OPERATION

OPERATOR OPENS BALL VALVE BY HAND AND ALLOWS AIR FROM TANK TO FLOW OUT TO SMALL AIR MOTOR WHICH RUNS A BOOSTER COMPRESSOR.

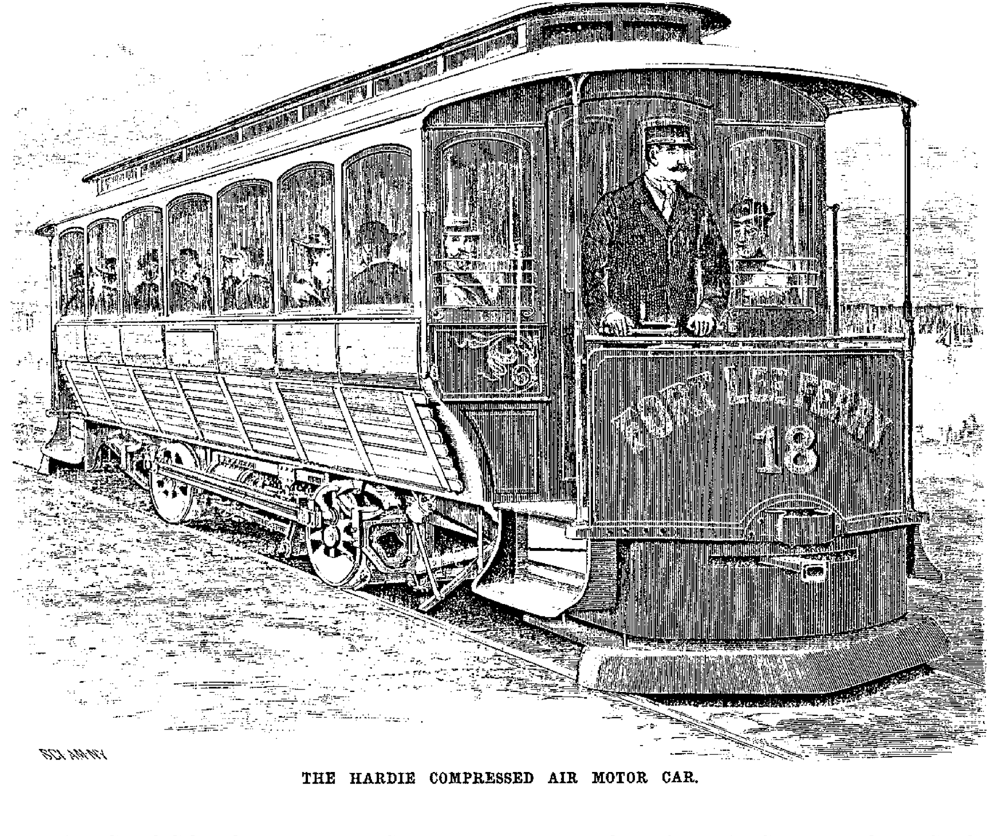
THE BOOSTER TAKES SOME OF THE AIR FROM THE DEPARTURE PORT AND INCREASES ITS PRESSURE. ON ITS WAY BACK INTO THE INTAKE PORT OF THE TANK, THIS HIGH PRESSURE AIR (200-2000 PSIG, AS NEEDED) ENTERS THE DRIVE THROAT OF THE JET PUMP, A NOZZLE THAT IS DESIGNED TO CONVERT ALL OF THE DRIVE AIR'S PRESSURE INTO VELOCITY.

THE RESULTING KINETIC ENERGY AVAILABLE FROM THE JET PUMP NOZZLE DRIVE AIR IS USED TO ENTRAIN (DRAG IN) LARGE QUANTITIES OF ATMOSPHERE TO THE INTERIOR OF THE TANK. RESIDUAL ENERGY IN BOOSTED DRIVE AIR NOT NEEDED OR USED TO ENTRAIN ATMOSPHERE RE-ENTERS TANK AND IS CONSERVED.

TESTING GOALS

IF NO ATMOSPHERE IS ENTRAINED, PROXIMITY TO SUCCESS IS DETERMINED BY PRESSURE GAUGES CONNECTED TO THE THREE SPACES DEFINED IN THE TANK INTAKE LINE BY THE DOUBLE CHECK VALVE DISCHARGING INTO CENTER OF TANK, JUST UPSTREAM FROM JET PUMP DRIVE NOZZLE. THIS DEVICE IS THE EQUALIZER AND IS WHERE THE ATMOSPHERE ENTERS THE LOOP OF BOOSTER AIR. PRESSURE IN THE EQUALIZER DISCHARGE TAILPIPE LOWER THAN MAIN TANK PRESSURE INDICATES THE RESISTANCE THAT MUST BE OVERCOME BY INCOMING ATMOSPHERE. IDEAL JET PUMP SUCTION PORT PRESSURE IS 5 PSIG OR LESS FOR SELF-FUELING OPERATION.

IF ENOUGH ATMOSPHERE IS ENTRAINED AT A LOW ENOUGH COST IN STORED POWER, EXTRA COMPRESSED AIR BECOMES AVAILABLE FOR USE IN UNRELATED ACCESSORY WORK SUCH AS RUNNING A CAR, PUMP, LAWNMOWER, OR GENERATOR. DRIVE TANK COLLECTS AIR RELEASED FROM EQUALIZER TANK BY SAFETY VALVE, AND THIS AIR IS AVAILABLE FOR EXTRACURRICULAR USE.



THE HARDIE COMPRESSED AIR MOTOR CAR.

THE PNEUMATIC LOCOMOTIVE INDUSTRY THRIVED FOR 50 YEARS. THE ONE SHOWN HERE WAS USED FOR METROPLITAN TRANSIT IN NEW YORK CITY BEFORE THE TURN OF THE CENTURY. ITS GREATEST SUPPORTER WAS GENERAL HERMAN HAUPT, A CIVIL ENGINEER WHO RAN THE RAILROADS DURING THE CIVIL WAR.

Interview with George Heaton, October 31, 1980

George told me he'd take me aside after he'd had a little more time to think. When he finally beckoned me to join him in another room, I was paying attention. Though I wondered if this might be a prank or a misunderstanding, he hadn't been talking long before I had my notebook out.

And there was no reason to think that either Maria or her husband was a prankster. I knew from spending long hours with Maria at the shop where we both worked that she was an intelligent, conservative working mother and the wife of a friendly and generous man of many accomplishments. As he began to reminisce, George was telling me that in 1969 he'd been the vice-president of the California Fuel Dealers Association. In that role, he'd testified before a legislative committee concerning the environmental dangers involved in the use of the catalytic converters needed with unleaded gas. He also warned me about compressor explosions, saying that one drop of oil contamination in an oxygen compressor could blow up a whole building. Unlike oxygen, compressed air isn't explosive, but he wanted to impress me with the seriousness of taking safety precautions when working with pressure equipment.

George told me that around 1949, he and a friend converted some motorcycles and cars to run on compressed air. They converted the existing gas engines to run on air, with several modifications that George described.

The cars they built "worked like perpetual motion machines." His wording seemed to imply that the car seldom or never ran out of air, that this is considered impossible by those who should know, that it obviously is not impossible because George has done it, and that it's still considered impossible by those who should know. The key to doing the impossible was to put low pressure air into a high pressure tank, without having to compress the air first, that is, without having to force it in against the resistance of the pressure in the tank. This allowed the use of small air pumps running off the car's motion to keep the tank full, while the engine ran off compressed air leaving the other end of the tank. George didn't remember exactly how to get low pressure air into a high pressure tank, but he thought a good compressor man could probably figure it out. He did recall that the air entered the tank in a stream of quick spurts, or pulses, and he thought it might have entered the tank "at an angle or something."

George and his friend "weren't engineers enough to know what pressures to use," and as a result they had trouble with their engines blowing up. That didn't keep them from driving their air cars across the country several times. The last time George was driving his air car across Nevada, a piston blew out the top of the engine, through the hood, and up into the sky, where it disappeared from sight. Since gasoline was so cheap at that time, the hassles and hazards involved in building experimental cars outweighed the disadvantages of buying gasoline, and they built no more air cars.

George suggested I get around the problem of converting existing engines to run on air, by using a 100 horsepower turbine air motor, which he said would weigh only 25 pounds. My later research turned up the 51A turbine air motor made by the Tech Development Co. of Dayton, Ohio.

Interview with Bob Neal's Son

Luther Rangely, October 14, 1988

(Floyd Neal started right in describing the engine hardware in some detail. I got my tape recorder hooked up while he was talking, and steered the conversation toward the equalizer, or "special valve" in the tank.)

LR: Did you see the inside of the tank where that one special valve was?
FN: Now he had a special valve where he could load the tank with very little pressure. That was a--the valve looked like an extremely skinny, long plumb bob. That's about all I can remember, like I say, I was just a small boy.
LR: How old do you think you were? Maybe 16 or so?
FN: Oh no, I was younger than that. Oh, probably maybe seven or eight years old, and probably the last I had anything to do with it, 'cause I was out going to school, probably maybe 13 to 15. But I couldn't really give you any good detail.
LR: Do you know about how long or how big the tank was?
FN: Well the storage tank was a streetcar tank. If I remember right they were probably about 16 inches by probably 4 feet.
LR: Pretty big tank, huh?
FN: Yes, the reason he used that, it was available. You probably wouldn't have to have that big a tank. As far as that goes, it was actually just to start it with. 'Cause then you see it starts producing air on its own.
LR: Do you know what the principle is of being able to get the low pressure air into the tank?
FN: That was, he felt, the valve. It was a type of valve that--it was a double valve of some sort.
LR: Double check valve according to the patent.
FN: Yeah, and you could load the tank with a lot less pressure than was in the tank.
LR: Did he ever talk about water hammer or pulsejets?
FN: No...
LR: You know when your water pipes start buzzing, vibrating in the wall, that kind of principle is what makes pulsejets work, and I was thinking possibly it was similar to that.
FN: I couldn't really tell you. Have you come up with anything that you're working on?
LR: Well no, I'm just a researcher, and this is so far the only patent I've found that actually said what it was trying to do. It doesn't say what the working principle is but I think I've figured it out. I think if you make the air vibrate, then it organizes itself into high pressure zones and vacuum zones, and the vacuum waves can be used to let that low pressure air in. So it's kinda like a ram pump, and pulsejets and other wave-type machines that work on causing the fluid to vibrate and make waves. So that's what I think it is, it seems to make sense to me, and that's what my research seems to lead to.
FN: Well it's important to get that research. Have you actually developed any kind of engine?

LR: No, what I've got is, I've built the tank and I put the two check valves inside the tank, sorta like the way it looked in the patent, and I've got an air motor running a rotary compressor, to put the low pressure air in. And I'm getting low pressure air into the tank all right--

FN: About what kind of pressure?

LR: About maybe ten pounds--I can get it in at two pounds but if you run the air motor faster it pumps it up to a higher pressure. It's still going in at a much lower pressure than what's in the tank. So I think it's working, and I think having the compression and engine cylinders on the same crankshaft is the secret. That's why I'm running the compressor direct with the air motor, so they're going at the same speed.

FN: Sure.

LR: Yeah. I think that's your dad's trick is to have the pulsations entering and leaving the tank at the same time so you just have that very clear, distinct wave in it.

FN: Does Mr. M. have a picture of the engine?

LR: Well he says he's got an article but he says it's off in a box somewhere and he doesn't even know where to start looking for it.

FN: I thought he might have a photograph. Of course you're trying to discover something that's altogether different?

LR: Well, yeah, I'm mainly sort of an air car advocate you might say. I'm exploring the whole area and I'm looking at all the different systems I can find, and so far I think this is the best, and I'm concentrating on it. And hopefully I'd like to build an air car that uses this principle--is the patent owned by someone now?

FN: No, you know, it's run out. It's public domain. I was thinking about, oh a few years back, renewing it, but then it wasn't right for me and I just didn't deal with it and now I have no interest in it.

LR: Well it seems to me like it sort of works similar to a perpetual motion machine, which is supposed to be impossible.

FN: You know, when he patented that thing--he had trouble patenting it. Because they notified him and told him that the United States Patent Office was not interested in perpetual motion. And he fired a letter back to them and told them he wasn't either, he just wanted to patent an engine that was functional. And as a matter of fact he got busy and made a little--a small prototype--a hand carried model. And as a matter of fact I went with him to Washington when I was a little boy, and he put it up on Garrett Whiteside's desk--he was top man at the time--started the thing up, and he called in the investigative men, and they had to issue him a patent. You can't argue with a functioning engine, right?

LR: That's right. Well that's great. So the little model had the tank and the valve and the--

FN: Yes, it had everything and he could actually carry it in.

LR: Well that's pretty good because they say in no uncertain terms that they won't grant one, and there's so many air car

patents that--
FN: Yes. Don't ever mention perpetual motion.
LR: Right. Have you ever heard of anybody else that's done something like this with air?
FN: No, I heard there was someone in the South, a couple years ago, but I don't know if it was just rumors or what. But I didn't really know about it.
LR: What was your dad's relationship with Mr. M.? They were corresponding with each other? Or they were friends?
FN: Well, I don't really know how they got--oh, I know what it--my dad's sister, my aunt, and her husband were living in California somewhere, and I think they were sitting in a city park or something, and the conversation just came up, during the conversation, and my uncle said, well, his brother-in-law was--oh, it was "odd things"--but he said his brother-in-law was working on an engine that ran on air. And Mr. M. heard that, and got interested, and got his address and everything and came down. I remember when he came down. That's how that started. I think that was about '45.
LR: Well I've been working on this research for nine years and someone introduced me to Mr. M. over the phone. We had a three-way phone conversation and he started talking about this and we tracked down the patent. And I thought about it for about a year and a half before I figured out this wave principle for how it might work, and got some research to back it up. I think we're doing pretty good. Would you like to be on my mailing list in case we get something going and put out a newsletter?
FN: Yeah. You know, there was another fellow that was interested in this; that was William Lear. He was kind of interested in seeing an air engine a couple of years ago. He came down to see me a number of times also. And that was over this valve--in the tank.
LR: Did he look at it?
FN: Well, I didn't have it. He came down to my place.
LR: Oh. He's sort of like me, he was trying to get information from you.
FN: Right. And I don't know, he did come up with some sort of an engine--steam, though, I think--and busses in L.A., years ago, and also in police cars. But I don't know.
LR: Was he adapting your father's invention?
FN: Well, at the time, he didn't know if he was gonna go air or steam.
LR: Mr. M. says you had to stop making this or stop developing it because somebody came from some government or something.
FN: That was during the war years?
LR: Uh-huh.
FN: Yes, as a matter of fact, my sister was even kidnapped over it. Germany wanted it real bad. They tried to buy it. And of course my dad didn't do business with foreign powers or the enemy. Then they tried it their way. And they threatened him and said they'd have his family members killed off one at a time. What happened is he dismantled it, and scattered the parts all over the countryside. They just

literally scared the old boy to death.

LR: That was the Germans, huh?

FN: Uh-huh.

LR: That's pretty bad.

FN: Yeah. Poor timing for him.

LR: That was right after he got it patented?

FN: Yes. His first engine was a lot bigger. The first one was fourteen cylinders--air compressors--it was a big "V". Then he decided that was too much, and then his last one was--he called it the "Model 39"--it was just half a block. It looked very similar, and the same size, as an old straight-8 Buick, because the hangers and everything would drop right in on a straight-8 Buick. Because that's what he had at the time, and he wanted that to drop in there, and use the same drive line and everything. The engine actually looked like a letter T.

LR: A letter T?

FN: A letter T. Just straight like an old straight-8 and it was a "V" out front where the two pullers were, set a little off to the side. The crank on that was perfectly round. The pullers had a little larger throw. The engine was basically the same thing cut in two from the original patent. Seven compressors is like fourteen, working both up and down. I remember it, he hooked it up to a machine lathe and had it running that. It worked!

LR: All right. Well I think it'll work because I'm partially showing it myself, a pretty crude, rigged-up system.

FN: Well, have you actually produced an engine?

LR: Well, not an engine, I'm just putting components together, I've used an air motor and I'm running a compressor with it, and I'm having that compressor put the air back into the tank. It's not putting enough back in the tank yet. Does his air motor cylinder--did they use the air pretty efficiently or did they let the air come in through the whole stroke and then exhaust it?

FN: Well, evidently it was efficient but he had no way of using the same air. His engine, I believe it would fill the tank I believe to 140 or 150 pounds and then the excess would escape.

LR: And then the exhaust from the engine just exhausted, right? It wasn't recaptured?

FN: Yeah, just like a regular combustion engine.

LR: Was the safety valve letting air out all the time or only when it was idling?

FN: Well, it maintained that pressure. And if I remember right, I could hear the air leaving all the time, so I think it was producing quite a bit more even under load.

LR: So even when it was running, the safety valve was letting out a kind of a regular spurt?

FN: Yeah. Uh-huh. It made a hissing noise because he didn't muffle it. He just turned it loose.

LR: That could be a key. If the safety valve was letting air discharge all the time, then that's important because it could be causing the pulsejet effect. Sudden discharge like that--

when you suddenly let a burst of air out through your safety valve--can create a vacuum inside the tank.

FN: Sure. That's what it is, it's one of the features. That's what he said, "The valve is the feature." And like I said, as a small boy, I was thinking of other things.

LR: Right. Well, those are pretty much the questions I had for you, I really thank you for taking time with me--

FN: Well, I wish I could help you more, I just can't remember the finer details.

LR: Well, that's fine, I think that the patent's pretty complete, but they just left the working principle out. It's probably just patent lawyers' tactics, you know.

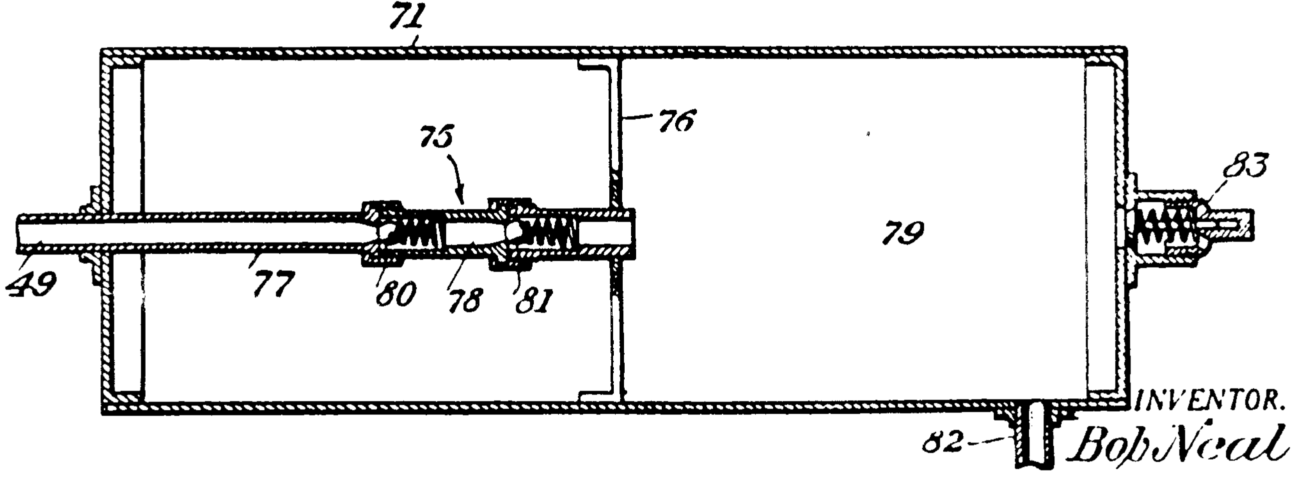
FN: Yeah, sure. That's the legal people for you.

LR: Yeah. All right, Mr. Neal, I appreciate your taking time with me.

FN: Well I appreciate talking to you. Hope it works out.

CONSTRUCTION

Fig.6.



Bob Neal's U.S. Patent #2,030,759 shows an equalizer (75) that lets the compression cylinders pump low pressure air into a high pressure tank (71) without compressing it to get it in. Possibly the Maxwell's Demon that physicists have been debating for over a hundred years, the Neal Equalizer may decrease the cost of compressing air to nearly zero.

Feb. 11, 1936. B. NEAL 2,030,759

COMPRESSOR UNIT

Filed Jan. 9, 1934 3 Sheets-Sheet 1

Fig. 1.

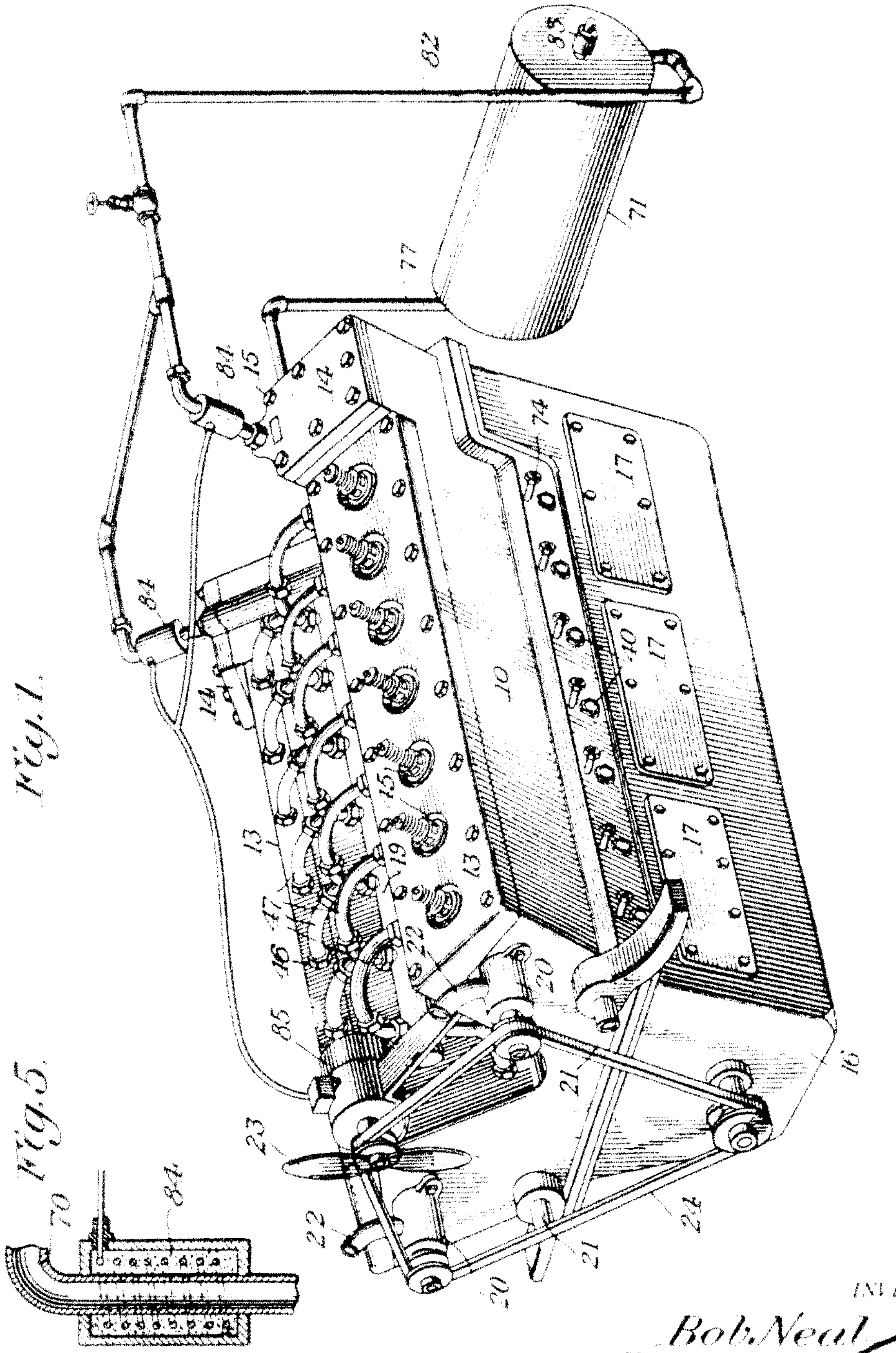


Fig. 5.

INVENTOR.

Bob Neal

BY

Paul O. Parker

ATTORNEY.

Feb. 11, 1936.

B. NEAL

2,030,759

COMPRESSOR UNIT

Filed Jan. 9, 1934

3 Sheets-Sheet 2

Fig. 2.

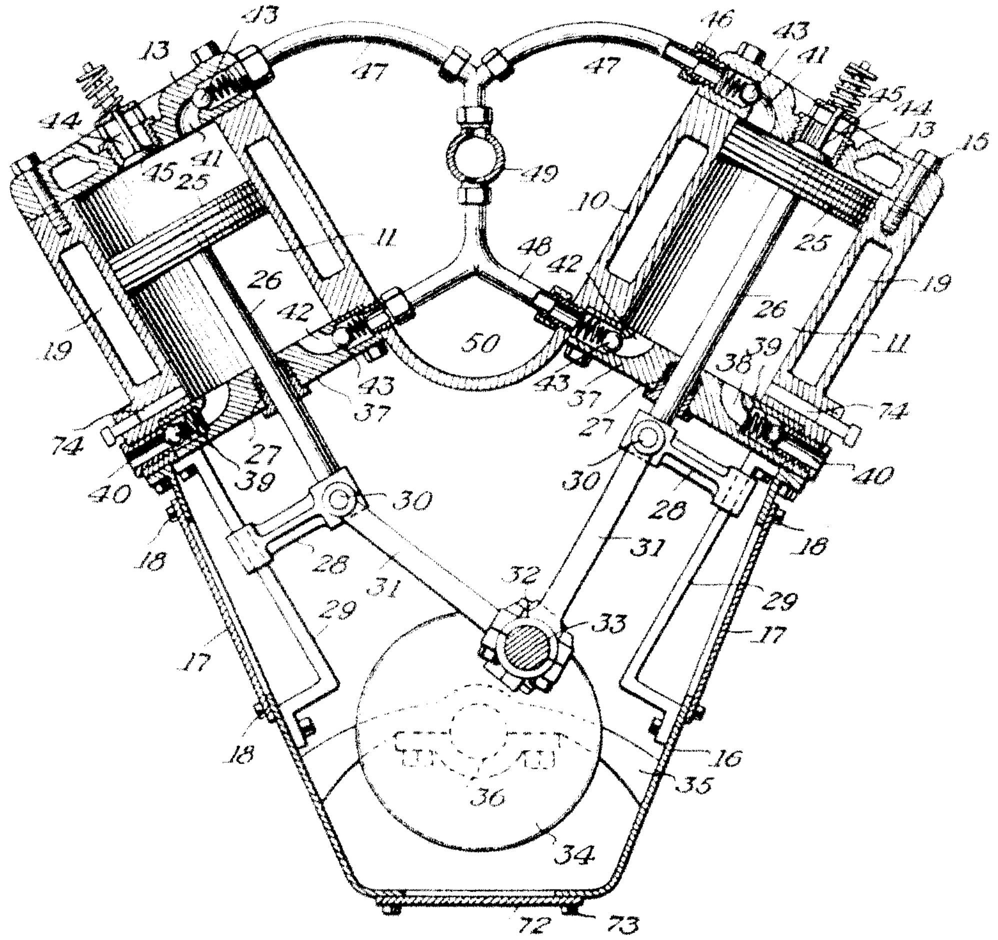
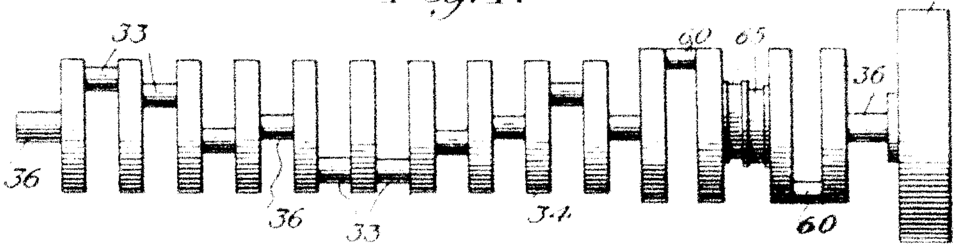


Fig. 4.



INVENTOR.

Bob Neal

BY

ATTORNEY

Feb. 11, 1936. B. NEAL **2,030,759**

COMPRESSOR UNIT

Filed Jan. 9, 1934 3 Sheets-Sheet 3

Fig.3.

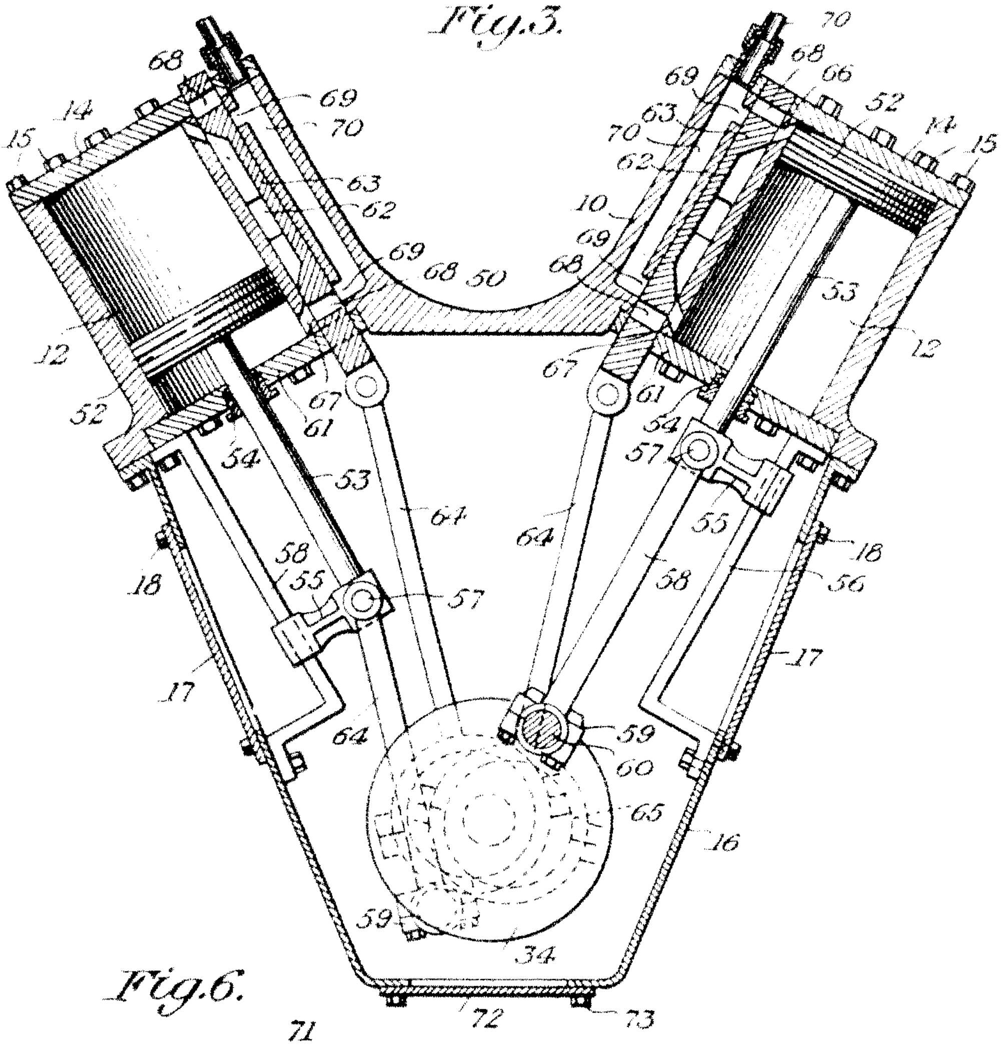
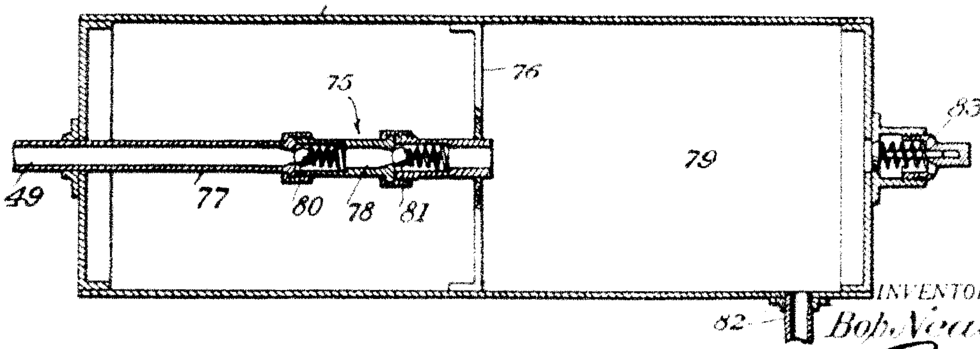


Fig.6.



INVENTOR.
Bob Neal
BY
ATTORNEY.

Patented Feb. 11, 1936

UNITED STATES PATENT OFFICE

2,030,759

COMPRESSOR UNIT

Bob Neal, Arkadelphia, Ark.

Application January 9, 1934, Serial No. 705,964

1 Claim. (Cl. 230—187)

The invention relates to a compressor construction, and more particularly to a combination fluid operated engine and compressor.

The primary object of the invention is the provision of a compressor of this character, wherein there is arranged an automatically counter balanced crank shaft and fluid equalizers within a storage tank, which makes it possible for the said engine to operate on constant reserve tank pressure so as to actuate additional equipment, the pistons for the engine being also automatically balanced and suspended when the said engine is in operation.

Another object of the invention is the provision of an engine of this character, wherein the same is operated from air under pressure, the said air being supplied by compressors, these being in bank with the engine construction.

A further object of the invention is the provision of an engine of this character, wherein the same is of novel construction, as the engine proper and the compressors are operated from the same crank shaft which is of the automatically balanced type, so that high efficiency is attained.

A still further object of the invention is the provision of an engine of this character, which is comparatively simple in construction, thoroughly reliable and efficient in its operation, strong, durable, and inexpensive to manufacture.

With these and other objects in view the invention consists in the features of construction, combination and arrangement of parts as will be hereinafter more fully described, illustrated in the accompanying drawings, which disclose the preferred embodiment of the invention, and pointed out in the claim hereunto appended.

In the accompanying drawings:

Figure 1 is a perspective view of the engine constructed in accordance with the invention.

Figure 2 is a vertical transverse sectional view through the compressor part of the engine.

Figure 3 is a vertical sectional view through the power part of the engine.

Figure 4 is a detail elevation of the crank shaft of the engine.

Figure 5 is an enlarged sectional view through one of the electric heaters for the engine.

Figure 6 is a vertical longitudinal sectional view through the air storage tank including the equalizer.

Similar reference characters indicate corresponding parts throughout the several views in the drawings.

Referring to the drawings in detail the engine in its entirety comprises a cylinder block **10** having formed therein the series of compressor cylinders **11** and the power cylinders **12**, respectively, the block **10** being of the V-type and closing the upper ends of said cylinders are the removable heads **13** and **14**, respectively, which are secured in place by head bolts **15**, as is conventional. Beneath the block **10** is the crank case **16**, which at opposite sides carries the detachable plates **17**, these being held in place by fasteners **18** and such plates are seated so as to be leak proof. The block **10** is chambered to provide a water jacket **19** about the cylinders, while at the forward end of the said block are water pumps **20** circulating water through the inlet pipe **21** which leads into the jacket and letting said water out therefrom through the outlet pipe **22** leading from said water jacket. Next to the pumps **20** is a fan **23** operated from a belt **24** which also drives the pumps.

Working within the cylinders **11** are the reciprocating pistons **25**, their rods **26** being slidable through packing glands **27** and fixed to crossheads **28**, which are slidably mounted upon guides **29** secured within the crank case **16** to opposite side walls thereof. These crossheads **28** are fitted with wrist pins **30** pivotally connecting therewith the connecting rods **31** which by the bearings **32** are engaged with their cranks **33** of a counter balanced crank shaft **34**, which is mounted in supports **35** arranged in the crank case **16**, the shaft being supplied with the required bearings **36**.

The inner ends of the cylinders **11** are fitted with inner end heads **37**, which are provided with air intake ports **38**, these being fitted with spring ball inlet checks **39**, the air having admission through passages **40** opening exteriorly of the block **10**. The glands **27** are associated with the heads **37**.

The heads **13** and **37** are provided with the compressed air outlets **41** and **42**, respectively, these being fitted with spring ball checks **43**, the heads **13** being also provided with the central air inlets **44**, which are fitted with spring checks **45**. By couplings **46** are attached to the air outlets **41** and **42** the outlet feed pipes **47** and **48**, respectively, these leading to a main conduit **49** located in the center channel **50** in said block **10**.

At the rear end of the block **10** and on the shaft **34** is the fly wheel **51**, this being of conventional type.

Working within the cylinders **12** are pistons **52**, their rods **53** sliding through packing glands **54** and fixed in crossheads **55** slidably mounted upon guides **56** which are secured within the crank case 15 at opposite side walls thereof. The crossheads 55 carry wrist pins 57 connecting therewith connecting rods 58, these being engaged by bearings 59 with their respective cranks 60 of the crank shaft 34, the inner ends of the cylinders 12 being also closed by inner heads 61 with which are associated the glands 54.

On the cylinders 12 are slide valve chests 62 in which are the slide valves 63, these being operated by throw rods 64 actuated by cams 65 and such valves controlling the air admission and exhaust of air to and from the cylinders 12 through the ports 66 and 67, and these valves 63 are provided with the ports 68 for the delivery of air under pressure from the inlet passages 69 common to a lead 70 from a compressed air storage tank 71.

The bottom of the crank case 16 is fitted with a removable plate 72 which is secured in place by fasteners 73, and when this plate is removed access can be had to the crank shaft 34 and the bearings for the engine, as well as other parts within said crank case, as should be obvious.

Leading into the cylinders 11 are the passages 74 of a lubricating system (not shown).

The storage tank 71 for the compressed air includes therein a double check discharge nozzle 75, this being supported by a member 76 and leading to this equalizer is an air inlet pipe 77 which has the communication 78 with the chamber 79 formed by said tank. In the equalizer 75 are the spaced spring ball checks 80 and 81, respectively, one being for the inlet side and the other for the exhaust or outlet side of said equalizer. This pipe 77 is connected with the main conduit 49, while a pipe 82 is connected with the leads 70, the tank being also fitted with an automatic relief valve 83 of any approved type.

About the pipes 70 for the passages 69 are the electric heating units 84 which are for the purpose of heating the air under pressure above a freezing temperature when delivered from the tank 71 to the cylinders 12.

Supported on the block 10 is an electric generator 85 which is driven from the shaft 34 through a belt 24 and this generator is included in an electric circuit which also has the heaters 84 so that these will operate from current furnished by said generator.

The storage tank 71 with the equalizer is so constructed that it is possible to pump air into the said tank with a tank pressure of two hundred pounds, while the compressors are only pumping against fifteen pounds or atmospheric pressure. Outside air pressure source can be coupled with the tank to augment that pressure derived from the cylinders 11 of the engine.

What is claimed is:

In a structure of the kind described, a V-shaped cylinder block provided with upwardly divergent cylinders, end heads fitted to said cylinders at opposite ends thereof, each head having valved inlets and outlets, a main outlet lead between the cylinders of the block for a storage tank and having lateral branches to the outlets at the inner sides of said heads, one inlet being located at the center of each head at the outer ends of said cylinders while the remaining inlets are at the outer sides of the heads at the inner ends of said cylinders, a substantially V-shaped crank case fitted to the block beneath the cylinders, a counterbalanced crank shaft journaled in the crank case, pistons operating in the cylinders and having rods extended into the crank case, crosshead guides fitted to the sides of said case interiorly thereof, crossheads connecting the rods with the guides and slidable on the same and connecting rods operated by the crank shaft and pivoted to the crossheads for reciprocation of the pistons.

BOB NEAL.

Chain of Invention

From the First Engine to Bob Neal's Equalizer

by Scott Robertson, 1989

In 1670, Christiaan Huyghens devised the first-ever engine, made up of a cannonball acting as a piston in a vertical cylinder. An explosion of gunpowder under the ball would raise it in the vertical cylinder, then it would fall back down the cylinder under its own weight. Huyghens' engine was able to pump water by the suddenness of the motion imparted to the cylinder's contents by the explosion. Behind the high pressure explosion pushing the ball up the cylinder, there was an implosion, a sudden and short-lived partial vacuum or rarefaction wave, in the cyinder. Because of the subatmospheric pressure, or depression, in the cylinder, the atmosphere added its own energy to this engine cycle to force the next cylinderful of water into the rarefied space under the ball. Then the ball fell back down the cylinder, pumping the water back out. Each explosion would "scavenge" the cylinder by clearing it of any water left over from the last cycle, by means of a high pressure blast or compression wave.

Huyghens' basic discovery--the depression left inside a closed cylinder after a sudden outward pulse--was the working principle that Thomas Savery used in taking the engine to the next stage in its development. The alternating wave in the fluid to be pumped comprised a fluid piston that did the pumping. Pulses of steam alternately scavenged the two chambers of Savery's engine, and each chamber would automatically refill with water because of the depression left behind each scavenging pulse. Savery's engine had no moving parts except valves. The mass exit of the chamber's contents left a depression that induced the next chamberful of water, and a pulse of steam pumped this water out. The pulsometer pump, which was manufactured from 1876 to at least 1938, used the same principle.

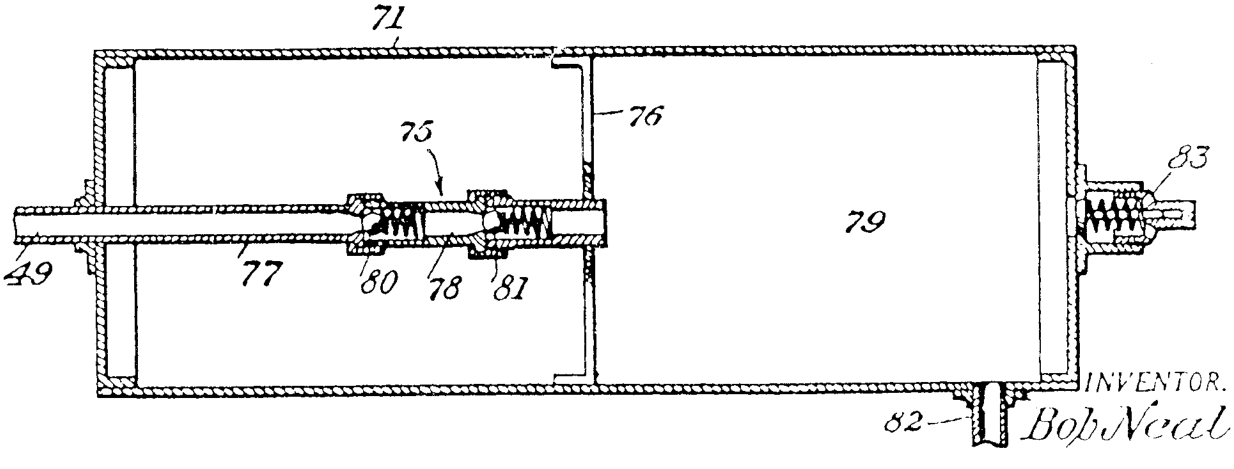
Newcomen's and Watt's cumbersome piston engines, laden wth expensive moving parts, took precedence in the developing engine industry over the simpler fluid piston principles that Huyghens and Savery pioneered. This was, in effect, an early version of engineered obsolescence.

Bob Neal's patented Compressor Unit is the only device I know of that can provide the essential hardware needed to run a self-fueling air car. Neal filed his patent in 1934; a trip to Washington with the working model secured him a patent in 1936; shortly thereafter he had to abandon the project due to harassment by the Nazis. The Nazis perfected the pulsejet engine in 1943; the French were developing their own pulsejet, which had no valves, at the same time. This pistonless piston engine--a tube containing a self-sustaining fluid piston--proved to be the most powerful engine for its size that was ever built. And none of the pulsejet's inherent defects, such as high noise levels or wasted residual energy in the exhaust, apply to Neal's "equalizer" since the equalizer is inside a tank full of compressed air.

Michel Kadenacy filed the first of his twelve U.S. patents in 1934, eight months after Bob Neal filed his patent. Kadenacy filed his French patents on August 1, 1933, a few months after Hitler took power. Kadenacy's system of scavenging and charging an engine cylinder is still the theory behind two-stroke engine tuning. The Kadenacy Effect could be called the Huyghens Effect, but Huyghens already had a principle named after him. The Huyghens Principle describes the tendency of each wavelet in a wave field to propagate its own wave field. The work of Huyghens became the foundation for the work of James Clerk Maxwell, the founder of mathematical prediction in theoretical physics, who predicted the discovery of overunity dynamic pressure exchangers (Maxwell's Demon) in his textbook THEORY OF HEAT in 1870.

Arkansas shoemaker Bob Neal's compressor unit is Maxwell's Demon reduced to hardware, and it's also the logical idealization of Huyghens' first-engine-in-history. The Neal Equalizer will one day be considered a refinement in engineering thought that paved the way for the age of sustainable technology to take hold in the 21st century.

Grass Valley, California
Groundhog Day, 1989



Resonance in Closed-ended Pipes

(written for this work by a retired mechanical engineer)

Considering only pressure changes--that is, compressions, C, and rarefactions, R--in a closed pipe whose length, L, is adjusted to be one-fourth the wavelength of the entering sound (Fig. 1), let us follow successive regions of C and R as they move into the pipe and are reflected from the ends, remembering that change of pressure phase occurs at the open but not at the closed end.

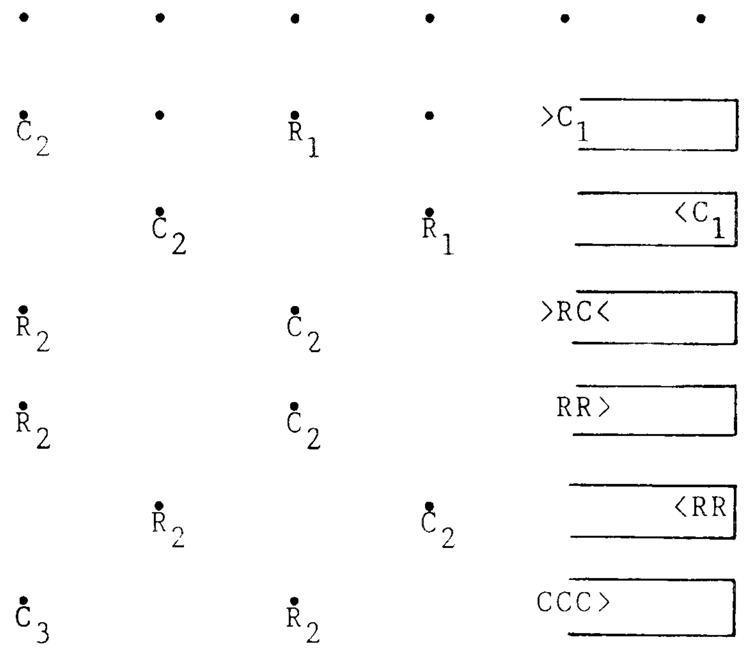


Fig. 1) resonance in 1/4- or 3/4-wave closed-end tubes

The compression wave C_1 enters the pipe, is reflected at the closed end, and reaches the open end just as R_1 is about to enter. As R_1 enters, C_1 is reflected back in the opposite phase as an R, and unites with entering R_1. The now amplified R_1 is reflected at the closed end, and on returning to the open end, is reflected as an amplified C, which is further amplified by uniting with entering C_2.

Thus the pressure disturbance traveling back and forth within the pipe is greatly augmented, and an intensity is built up within the pipe which is very much greater than that in a corresponding length of the incoming wave itself. In this way, a sound of such low intensity as to be inaudible outside the pipe becomes loud and clear on reinforcement be the resonance built up in an air column properly tuned to respond to it.

The same phenomenon occurs likewise if the closed pipe is three-quarters wavelengths of the incoming wave. Compression C_1 then unites with rarefaction R_2, and these again with C_3 and R_4; also, C_3 meets rarefaction R_3 and C_4. Similarly one gets resonance in a closed pipe if the length is any ODD number of quarter-wavelengths (see Fig. 1 again).

Resonance, closed pipe, for L = 1, 3, 5, 7,... $\frac{\lambda}{4}(2n-1)$

or for a given pipe length, response will occur for waves of length

$$\lambda_{1,\ 3,\ 5,\ 7,\ 9,\ldots} = 4L,\ \frac{4L}{3},\ \frac{4L}{5},\ \frac{4L}{7},\ \frac{4L}{9},\ \ldots$$

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CHAPTER THREE

Scavenging and Supercharging

General—The terms "scavenging" and "supercharging" refer to the means employed to introduce into engines the air required to support combustion of the fuel, plus a sufficient excess to control internal and exhaust temperatures. Moreover, the high mip of modern engines at rated power could not be obtained without means of charging the cylinders with large quantities of air.

The term "scavenging" is ordinarily associated with two-cycle engines and relates to the removal from the cylinder of products of combustion formed during the previous cycle. This is achieved by the products of combustion with a fresh charge of air, part of which constitutes the combustion air for the succeeding cycle. The term "scavenging" is not usually associated with a four-cycle, supercharged engine although part of the pressurized air provided by the turbocharger performs the same function.

Supercharging refers to the practice of filling the cylinder at the beginning of the compression stroke with air at a pressure *substantially* higher than atmospheric pressure. This makes it possible to burn a larger quantity of fuel per cycle and thus increases the horsepower output of the engine.

Scavenging air for two-cycle engines must be supplied at some pressure above atmospheric in order that it may perform its function. However, while the pressure at the beginning of the compression stroke may be *slightly* above atmospheric pressure because of resistance in the exhaust line, this is not considered to be supercharging.

Blowers or Compressors—Various types are used to supply the air for scavenging or supercharging. A normally aspirated, four-cycle engine having an exhaust stroke and a "suction" stroke, is inherently self-scavenging and self-charging, hence, requires no external device to supply its air. All two-cycle engines must be equipped with some kind of compressor to supply the scavenging air.

Two-cycle engines of the simplest type have check valves and ports in the crankcase so that the underside of the piston serves to draw air in on the upstroke and discharge it on the downstroke. This air is directed into the cylinder through ports which are uncovered by the piston as it approaches bottom center. Some crosshead engines use the underside of the piston as a compressor, but the capacity is limited except when used in conjunction with turbocharging.

Common types of compressors utilized for the foregoing purpose are discussed in the following paragraphs.

A. **Reciprocating Compressors**—These are commonly called "scavenging pumps." They are usually driven by a separate crank on the end of the engine crankshaft and are double-acting. On some of the larger engines they are built in tandem to obtain the required capacity.

B. **Oscillating Compressors**—These employ one or more vanes fixed radially to a shaft which oscillates 90 degrees or less, depending on size and number of vanes. Each vane oscillates between two fixed radial partitions in a cylindrically shaped housing with end covers and suction and discharge valves. When engine-driven, the compressor vane(s) is actuated from the crankshaft by a connecting rod linked to a lever on the oscillating shaft.

C. **Rotary Compressors**—This term refers to compressors of the positive displacement type of which there is a great variety. Only those commonly used in the supply of scavenging air for internal combustion engines will be discussed here.

1. **Roots-Type Blowers**—These consist of two rotors or impellers having close clearances mounted on parallel shafts in a casing having essentially the form of two intersecting cylinders. Each rotor has two or more lobes shaped to mesh between corresponding lobes on the other rotor. One rotor shaft is mechanically connected to the source of power and drives the other by means of a pair of timing gears fitted to the rotor shafts. These gears maintain the rotors in proper phase relation so that, while they are said to "mesh," the rotors are never in contact with each other but always have slight clearance. Air, trapped in the spaces between the rotor lobes and the casing, is swept around from suction to discharge as the rotors revolve. The meshing of the lobes effectively seals the space between the rotor axes against the backflow of air under discharge pressure. The construction is illustrated in Fig. 7. Roots-type blowers have no compression ratio inasmuch as there is no change in the volume from inlet to discharge. Air is pushed along and, due to resistance on the discharge side, pressure builds up.

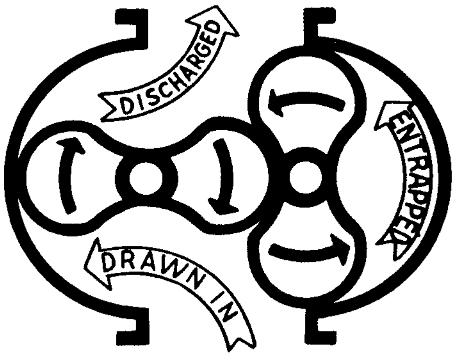

Fig. 7. Roots-type of Mechanical Blower.

2. **Axial-Flow Rotary Compressors**—This type consists of a suitable housing containing two conjugate helical rotors. The one on the driving shaft is the main rotor and usually has two lobes. The mating rotor usually has four lobes and is driven at half the speed of the main rotor by gears on the respective shafts which keep the rotors in phase and precisely maintain the small clearances between them. Similarly, there are small clearances between the rotors and the housing.

   The axial-flow compressor differs from the roots-type blower in that it has an inherent compression ratio. The air is drawn into one end of the housing at one side, carried around between the rotors and the housing, and compressed as it is forced toward the discharge end on the side opposite the suction. When compressors of large capacity are required, this type can be provided with three lobes on the main rotor and four on the other.

D. **Centrifugal Compressors**—These are used where large volumes of air must be compressed. The centrifugal compressor is a high speed machine that is particularly well suited to direct drive by an induction motor, if a-c power is available. If d-c must be used, lower speed motors and speed-increasing gears are required. The centrifugal compressor consists of a vaned wheel, called an impeller, which rotates in a volute shaped casing. As the name implies, centrifugal force is the basis of compression. The air enters the casing near the hub of the impeller and, through centrifugal force, is impelled into the casing volute at a high velocity. The volute is carefully designed to convert the velocity into pressure.

E. Turbocompressors—These are commonly called turbochargers. This type consists of a centrifugal compressor, driven by an exhaust-gas turbine. These two basic components are incorporated in a compact unit assembly, one or more of which may be used on a single engine, depending on capacity required and certain other considerations. They operate at higher speeds than the motor-driven centrifugal compressors, the larger sizes at 8000 to 10,000 rpm and the smaller ones at 30,000 rpm or higher. Turbocharging utilizes energy in the exhaust gases which otherwise would be wasted and thus improves engine efficiency. The speed of the turbocharger has no fixed ratio to the engine speed but will vary with the load, even with a constant-speed engine. Two basic methods of utilizing the energy in the exhaust gases are: (a) The pulse system; and (b) The constant-pressure system. The names are derived from the manner in which the turbine receives the exhaust gas. Both systems are applicable to two-cycle or to four-cycle engines. Descriptions of the two systems are:

1. **The Pulse Turbocharger**—The pulse system utilizes the kinetic energy in the exhaust gases as they leave the exhaust port or valve. Great care is taken in the design of the piping to the turbine to avoid dissipation of this energy. Since each firing of each cylinder results in an energy pulse, two or more separate pipes or groups of pipes are used, depending on the number of cylinders. These pipes are arranged so that the exhaust pressure wave of one cylinder does not interfere with the scavenging and charging air in another.

2. **The Constant-Pressure Turbocharger**—In the constant-pressure system, the exhaust manifold serves as a receiver. Thus, at any given load condition, the turbine receives a relatively constant pressure. The constant-pressure system is applicable to two-cycle and four-cycle engines and has an efficiency comparable to that of the pulse system but does not provide, by itself, the response to sudden applications of load that the pulse system does. In two-cycle engines either the constant pressure or the pulse system is frequently supplemented by other means of supplying air. This is discussed in greater detail later in this chapter.

Both two and four-cycle engines benefit from turbocharging as follows: (1) Power output is increased substantially; (2) Mechanical efficiency is higher; (3) Specific fuel consumption is less; (4) Lubricating oil economy is better; (5) Horsepower to weight and space ratio is improved.

Four-Cycle Engines—These are seldom built in substantial sizes without some degree of supercharging because the horsepower capacity of a turbocharged four-cycle engine can be two or more times the capacity of a naturally aspirated four-cycle engine of identical cylinder size and rotative speed. Upon starting a four-cycle turbocharged engine, the exhaust and intake strokes provide the engine with ample combustion air. With increasing load, the turbocharger is energized enough to provide the necessary air for supercharging.

Since the development of turbocharging, other methods of supercharging four-cycle engines have become obsolete.

The valve timing for a turbocharged four-cycle engine is arranged so that the exhaust and inlet valves are open concurrently at the end of the exhaust and beginning of the intake strokes. At this time they are said to "overlap." Air under pressure from the intake manifold can then pass through the combustion chamber to remove hot residual gases to perform a cooling function.

Two-Cycle Engines—These are always provided with cylinder wall ports which are uncovered by the piston(s). Such ports are used for admission of scavenging air, the release of exhaust gases, or both. In the latter case there are, of course, two series of ports, one connected to the scavenging header and the other to the exhaust header. Some engines use a combination of ports and valves. There is a great variety of these arrangements, the most common being as follows:

A. Loop Scavenging—This implies that the cylinder has both scavenging ports and exhaust ports. They are designed so that the scavenging air describes a loop toward the upper part of the cylinder to remove the residual gases and to recharge the cylinder with air. Two types are recognized, open-port and controlled-port scavenging.

1. Open-Port Scavenging

(a) Exhaust and Scavenging Ports on Opposite Sides—In the open-port system, the simplest case is shown in Fig. 8, with exhaust ports on one side and scavenging ports on the other. The exhaust ports extend higher on the cylinder wall so that the piston uncovers them first and releases the cylinder pressure before the scavenging ports open. The design of the scavenging ports, the shape of the piston, or both, serve to direct the air toward the upper part of the cylinder before it recurves toward the exhaust port, pushing the products of combustion before it and performing a cooling function. Since some air escapes

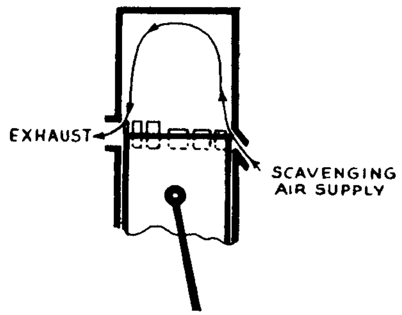

Fig. 8. Open-port Type of Loop Scavenging.

through the exhaust port, the capacity of the scavenging blower should be greater than the piston displacement.

(b) Exhaust and Scavenging Ports on Same Sides—One form of loop scavenging is shown in Fig. 9, in which both scavenging and exhaust ports are on the same side. The exhaust ports open first and, of course, close last. Air is directed across the cylinder, curving upward and making a complete loop before it reaches the exhaust port.

2. Controlled-Port System

(a) With Exhaust Nozzle Valve—This arrangement is similar to the open-port system shown in Fig. 8, except that a valve, fitted into the exhaust nozzle, closes when the exhaust scavenge period has ended. This prevents backflow of exhaust gases into the cylinder; thus it operates as a controlled-port system.

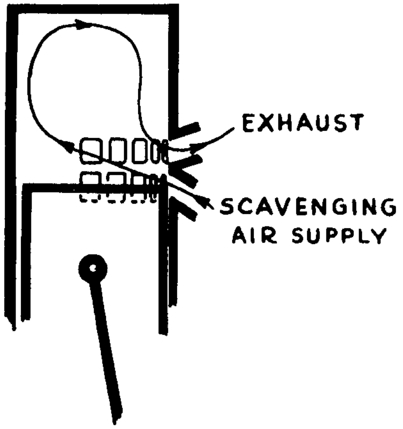

Fig. 9. Open-port Scavenging with Complete Loop.

(b) With Scavenging Nozzle Valve—Another form of controlled port is shown in Fig. 10. Here the exhaust ports open about the same time or a little ahead of the scavenging ports. Since the cylinder pressure at this point is higher than the scavenging air pressure, check valves must be provided outside the scavenging ports to prevent flow of gases into the scavenging air header.

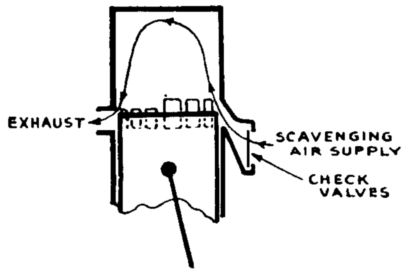

Fig. 10. Controlled-port Type of Loop Scavenging.

B. Uniflow Scavenging—This refers to a system in which the direction of airflow through an engine cylinder is from end to end. This is possible with either of the two arrangements described next.

1. **Poppet Valve Exhaust**—With this arrangement the exhaust valve or valves are located in the cylinder heads and are operated by cams and push rods. The scavenging ports are located at the opposite end of the cylinder from that of the exhaust. Exhaust valves can be timed to open before the scavenging ports, and close before they do, so that a cylinder charge pressure higher than otherwise possible can be obtained at beginning of compression. However, without turbochargers, it is not economical to raise this pressure beyond certain limits. With turbocharging, full advantage is taken of this feature.

2. **Opposed Pistons**—These provide an inherent uniflow system with exhaust ports uncovered by one piston and scavenging ports uncovered by the other when the pistons are at, or near, outer dead center. When the pistons are at inner dead center, the space between them forms

the combustion chamber. The two pistons in each cylinder may be connected to one crankshaft (see Fig. 11) or to separate crankshafts having a fixed phase relationship (see Fig. 12). By altering the phase angle of the cranks which control the opposing groups of pistons, it becomes possible to open the exhaust ports before the scavenging ports and to close the exhaust ports while the scavenging ports are still open. As in the case of the poppet exhaust valve system, advantage can be taken of this arrangement to retain a cylinder charge at higher pressure than otherwise would be possible.

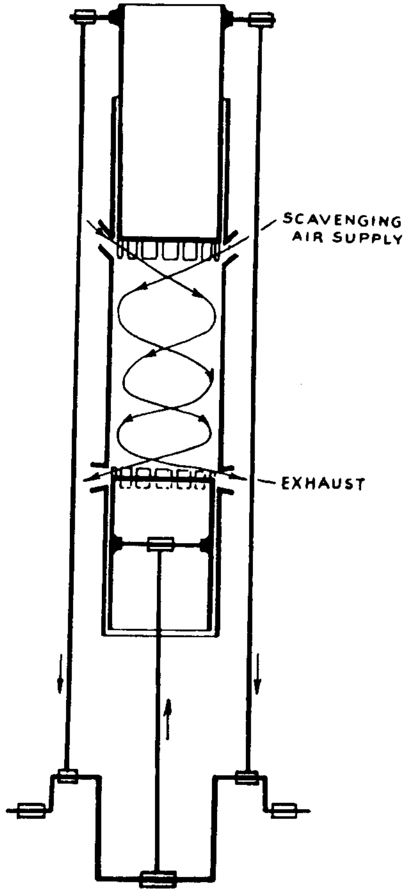

Fig. 11. Single Crankshaft Opposed Piston Uni-flow Scavenging System.

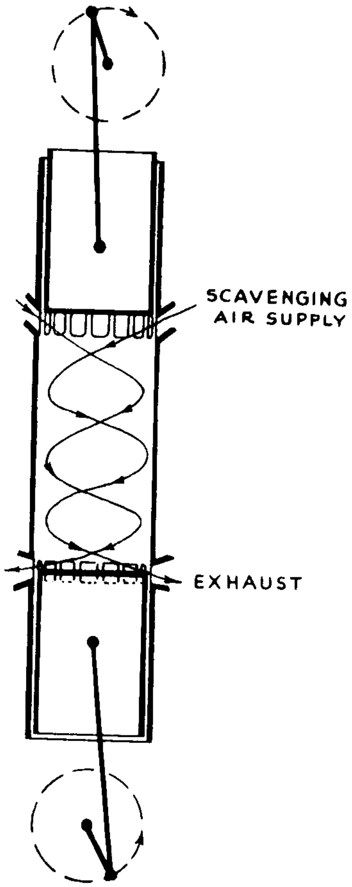

Fig. 12. Double Crankshaft Opposed Piston Uni-flow Scavenging System.

C. **Turbocharging**—The two-cycle turbocharged engine may require some means of obtaining combustion air during the starting period. Various means, whereby this is accomplished, are outlined in the following:

1. **Positive Displacement Scavenging Compressor in Series With Turbocharger**—With this arrangement, the positive displacement compressor, during the starting period, freely draws atmospheric air through the impeller of the turbocharger. As the load increases, the turbocharger builds up pressure and volume as its speed increases. On some crosshead engines, the undersides of the power pistons are used to provide combustion air for starting. These may be in series or in parallel with the turbocharger. In other engines, there is a relatively small reciprocating scavenging compressor driven by the crosshead of each cylinder and connected in series with the turbocharger. In normal operation, these compressors serve as a second stage of compression.

   Still other engines use a mechanically driven compressor in series with a turbocharger and with automatic provision for bypassing and unloading or declutching the mechanically driven blower when the load is such that the turbocharger is able to satisfy the air requirements of the engine.

   Another arrangement employs a turbocharger combined with a mechanically driven roots-type blower. Here, the mechanically driven blower is used for starting and partial-load operation. When the exhaust gases furnish enough energy, the turbocharger will develop, on the suction side of the blower, pressure equal to, or greater than, that on the discharge side. This causes the blower to unload its drive.

2. **Turbocharger With Overrunning Clutch**—Here the turbocharger, while mechanically driven, supplies air for starting and partial-load operation. The complete unit is driven through an overrunning clutch until the exhaust energy causes the turbine to increase speed and take over the driving function. At such time, the unit is disengaged from the mechanical drive of the overrunning clutch.

3. **Motor-Driven Centrifugal Compressor**—This is used to supply air during the starting period and to supplement the turbocharger. On some large engines with pulse-type turbocharging, such motor-driven compressors can be disconnected after the engine is started.

They are, therefore, arranged to cut in and out automatically.

4. **Air-Start Assist**—Air from the air-start tanks is sometimes used during the starting period to deliver air through jets to impart a rotating motion to the turbocompressor.

Cooling of Combustion Air—A turbocharged engine must have a compression ratio which will enable it to start under normal atmospheric temperature conditions. When starting, the turbocharger is inoperative, but when the engine comes up to load, the turbocharger is delivering air, not only at an elevated pressure but at a temperature corresponding approximately to that of adiabatic compression. The temperature of the air in the cylinder at the end of the compression stroke would then be much higher than necessary if some means were not employed to counteract the increased temperature. To overcome this disadvantage, the air is cooled after it leaves the turbocharger and thus permits safe operation of the engine at substantially higher loads than would otherwise be possible.

Internal Fire
C. Lyle Cummins Jr.
Carnot Press: Lake Oswego, Oregon, 1976

Chapter 1

Gunpowder and Steam

The internal-combustion engine began with the invention of the cannon. As a domesticated machine it did have the serious drawback that it threw away the piston on each power stroke. However, the potential for more peaceful applications of that explosive power released in the cannon's breech was not overlooked. In the late 1600's there were several serious attempts, using gunpowder as a fuel, to raise a piston in a vertical cylinder and then to let gravity and atmospheric pressure return it. Abbe Jean de Hautefeuille (1647-1724) proposed raising water by setting off a gunpowder charge in a chamber, releasing the resultant pressure, and then when the residue air had cooled, the vacuum thus formed would lift a quantity of water; he did not consider using a cylinder-piston mechanism[1] The power of a vacuum, along with the instruments to measure pressure in an evacuated chamber, had already been explained by the scientists Evangelista Torricelli (1608-47), Blaise Pascal (1623-62) and Otto von Guericke (1602-86).

A working piston in the cylinder of an engine employing heat was first constructed by Christiaan Huygens (1629-95). In 1673 he wrote of his experiments with gunpowder, calculating from these experiments that the equivalent of a 3,000 pound weight could be raised at least thirty feet by burning only one pound of gunpowder.[2] A free hand sketch sent to his brother of the proposed machine (Fig. 1-1) shows a piston D in a cylinder AB having ports E,E at the outer end of the cylinder. Attached to the port exits were flexible leather hoses F,F. When the piston reached the end of its upstroke due to the expanded gas, the ports were uncovered and the heated, spent charge was exhausted through the hose valves. Cylinder pressure then dropped to atmospheric, and with the cooling of the residue gas a vacuum was created. Outside air pressure pushed the piston inward, and on this return stroke a weight G was lifted. Providing a supply of fresh air for the next cycle was an inherent problem.

As late as 1729 there was a proposal for a jet propelled ship using a gunpowder engine. John Allen claimed in his patent,

> My method will be effected by forcing water or some other fluid through the stern or hinder part of a ship, at a convenient distance under the surface of the water, into the sea, by proper engines placed within the ship. Amongst the several and peculiar engines I have invented for this purpose is one of a very extraordinary nature, whose operation is owing to the explosion of gunpowder, I have found out a method of firing gunpowder *in clauso*, or in a confined place, whereby I can apply the whole force of it, which is inconceivably great, so as to communicate motion to a great variety of engines; . . . .[3]

Over sixty years passed before there was any further recorded work on an internal-combustion engine. Our story, therefore, takes a detour down the road of steam or external-combustion progress. A

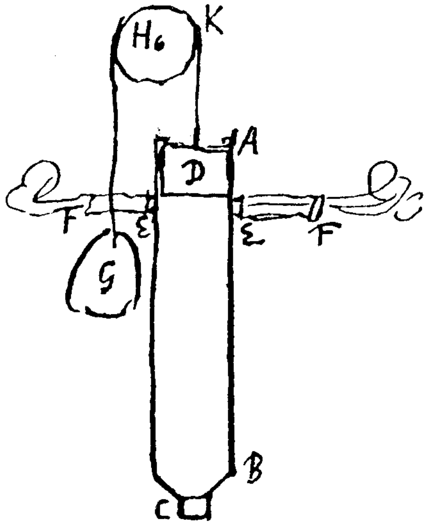

Fig. 1-1 Huygen's gunpowder engine, 1673. From a freehand sketch to his brother dated Sept. 22, 1673. (Klemm, *A History of Technology*, 1964)

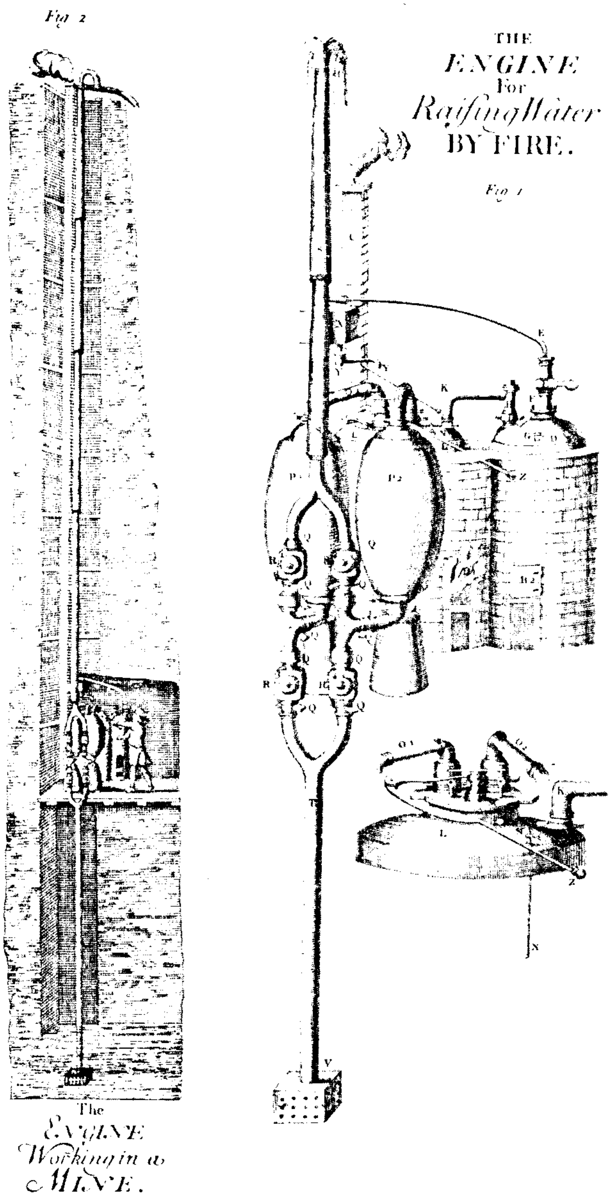

Fig. 1-3 Thomas Savery's "Miner's Friend." (The Science Museum, London)

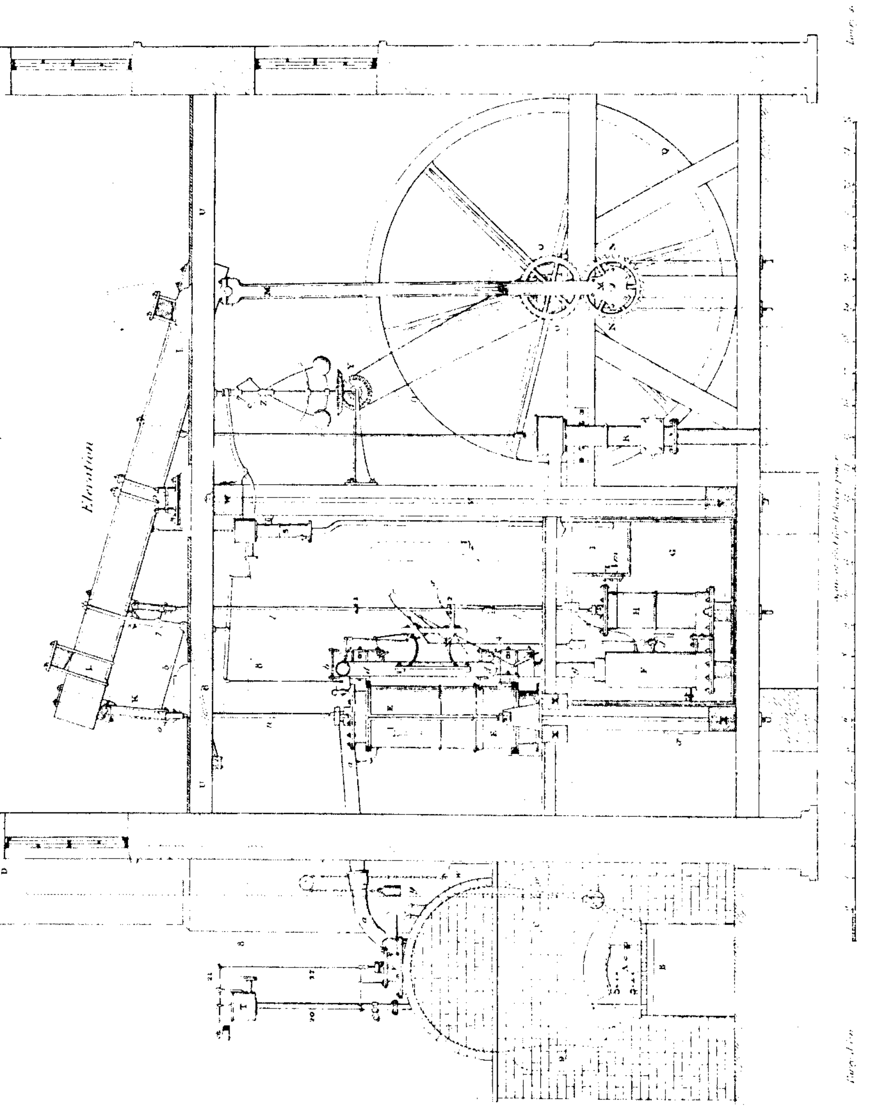

Fig. 1-8 Watt double-acting engine with sun and planet power transmitting gears, 1787. (The Science Museum, London)

look at the early evolution of steam power, while I-C development lay dormant, will help in understanding why the later non-steam pioneers made some of their conclusions and decisions. The 18th century steam era was also a time for the building of technology in metallurgy, machine tools and practical mechanism.

Denis Papin (1647-1712?), Huygens' gifted assistant during the gunpowder tests, soon turned his thoughts to the potential of steam power. Insight as to why is revealed in a letter he wrote in 1695 to a prospective client. A modern translation from a portion of this letter reads:

> As water has the property that it is converted by fire into steam . . . and can then be easily condensed again by cold, I thought that it should not be difficult to build engines in which, by means of moderate heat and the use of only a little water, that complete vacuum could be produced which had been sought in vain by the use of gunpowder.[4]

Papin's assumption that he could construct a working steam engine was based on a small model he had tested five years earlier. Here, a piston and rod were fitted in a two and a half inch diameter vertical tube. Steam generated by heating a little water in the closed tube raised the piston to the end of its stroke where it was latched. A vacuum was next formed by cooling the tube. When manually released, the piston was pushed downward by the atmosphere, as in Huygen's gunpowder engine. Unfortunately, the tube had to fulfill the triple function of boiler, power-cylinder and condenser. The idea was, therefore, quite impractical, but Papin was the first to capture the power of steam in a cylinder to move a piston. (As early as 1606 Giovanni Battista della Porta (1538-1615) had discussed how, by condensing steam in a closed chamber, a suction could be created for lifting water.)

Other contributions to the introduction of steam power by this exiled French Huguenot were the "digester," a form of pressure cooker or boiler, and a subsequent safety valve. It would be difficult to conjure a better example of invention following necessity than the addition of a pop-off valve to his boiler!

Thomas Savery

The fruit of Papin's work carried over to England, where as a member of the prestigious London Royal Society, a scientific study group chartered by Charles II, his efforts most likely planted a seed of an idea in the mind of a fellow member and inventor. This colleague, Thomas Savery (1650?-1715), subsequently developed and marketed about 1700 a steam-operated water pump called "The Miner's Friend." His pistonless engine (Fig. 1-2; Fig. 1-3, Plate) drew water into a pipe as a result of a vacuum formed by condensing steam in a chamber separate from a boiler, the lifted volume of water being trapped above a one-way or check valve. (The chamber might be termed a cylinder without a piston.) Steam was then readmitted from the boiler into the chamber, and with pressure now behind the trapped water, forced it upward past a second check valve into a discharge pipe. The condensing water and steam valves were hand operated. In addition to the basic inefficiency of alternately heating and cooling the pumping cylinder, the height water could be lifted (up to an estimated 150 feet) was directly dependent on available steam pressure. Consequently, to achieve the required lifts Savery was forced to employ pressures which often exceeded boiler, valve and joint capabilities. Imagine the Dantesque scene in a dark, hot mine shaft with the "Miner's Friend" hissing and leaking steam, the choking smoke from a firebox under the boiler, and a sweating, frightened attendant opening and closing valves. It was possible to

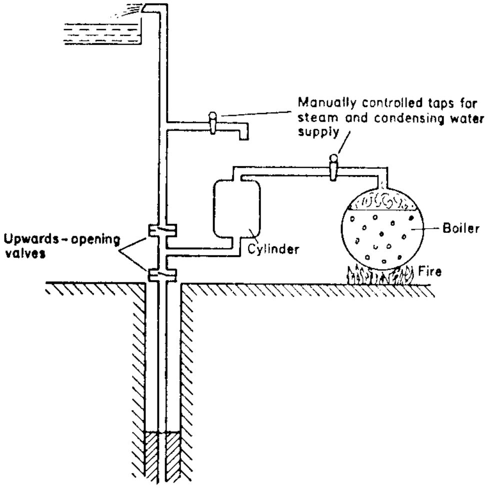
Manually controlled taps for steam and condensing water supply
Upwards-opening valves
Cylinder
Boiler
Fire

Fig. 1-2 Schematic diagram of Savery engine. (Cardwell, *From Watt to Clausius*, 1971)

operate the pump machinery up to five cycles per minute, but we have no reliable data on performance. Thermal efficiency is estimated to be much less than one-half percent.

Because of lift limitations and major safety problems, the Savery "fire engine" was soon relegated to the pumping of domestic water in a few of England's country houses and gardens. Here a lower steam pressure could be tolerated, and a servant could expect a longer life. Savery's concept had no lasting influence, even though it was a commercial success for a few years.

Savery's thinking was not confined to mines. He stated in his book extolling the virtues of the mine pumping engine: "I believe it may be very useful to Ships, but I dare not meddle with that matter; and eave it to the Judgment of those who are the best Judges of Maritain Affairs."[5] It is claimed in letters addressed to Leibnitz that Denis Papin, in 1707, ran one of Savery's engines in a boat on the Fulda River in Germany. Propulsion is assumed to be by the jet of exiting water from the pump. [illegible] first heat engine

Savery also enjoyed the protection of a generous English patent granted to him in 1698. The language of its preamble was not overly modest and likely reflected the thinking of the inventor-entrepreneur. His patent was for a

> . . . new invention of Raiseing of Water and occasioning Motion to all Sorts of Mill Work by the Impellent Force of Fire, which will be of great Vse and Advantage for Drayning Mines, Serveing Towns with Water, and for the Working of all Sorts of Mills where they have not the Benefitt of Water nor constant Windes.[6]

Originally issued for the normal period of fourteen years, the patent was extended by Parliament, for vague reasons today, an additional twenty-one years. (It is interesting to speculate as to the advantages of belonging to the London Royal Society at that time.) This patent was to assume far greater importance than just for Savery, as we shall see shortly.

Thomas Newcomen

Heat engine progress to 1700, Savery's work included, bordered on little more than novelty or scientific experimentation. Thomas Newcomen (1663-1729) changed this, and in turn altered the course of history. His intuitive genius plus years of trial and error determi-

A Short History of the Steam Engine

H. W. Dickinson

2nd Ed., London: Frank Cass & Co. Ltd, 1963

CHAPTER II

SAVERY AND HIS FIRE ENGINE

Savery employs pressure and suction for raising water—His patent and its extension—Great promise but failure for mining—Used for domestic supply—Papin makes a Savery engine—Blakey makes an automatic one.

IN the preceding chapter we have seen that many minds, scientific and practical, during the seventeenth century, were preoccupied with the problem of finding improved means of raising water from mines, and to a less extent for domestic purposes and irrigation. We have seen also that the field of research for such a power had been narrowed down to two avenues: the combustion of fuel to raise steam from water to a pressure above that of the atmosphere; and the explosion of small quantities of gunpowder to obtain like pressures. The problem of keeping such resultant high pressures within bounds, and the yet greater problem of repeating the operation as often as required with ease and economy, was as far as ever from being solved. It was the first of these avenues that offered the greatest hope of success, and it was this one that was followed; the second avenue was thus neglected, perhaps wisely, for it presented far greater difficulties than the first, and it took over a century before the pioneers cleared the way for the explosion, or as we now term it the internal combustion, engine. The scope of the present volume will not admit of following up the history of the latter branch of the heat engine.

The actual steps in the solution of the problem stated above were taken by Thomas Savery (1650?–1715), who was not only the first to produce a workable apparatus for raising water but was also the first to supply such apparatus to the public. The principle upon which Savery's apparatus worked can be stated briefly. Water is drawn up by suction from the height of, say 26–28 ft., i.e. that due to the pressure of the atmosphere, by the condensation of steam in a closed vessel; the column of water is there retained, and further steam is used to force the

water to a still higher level, suitable cocks turned by hand being provided for the purpose. It is to be observed that the apparatus is partly an atmospheric and partly an above-the-atmospheric pressure apparatus.

We are far from knowing as much as we should like to do about Savery personally. He was the younger son of a family of that name seated at Shilston, in the parish of Modbury, co. Devon, but whether born there or not, and when, is unknown. He is usually given the title of captain, but there is no record of a commission for a person of that name in any English regiment at that time. A Thomas Savery held a commission as "Ensign" in the army between 1688–97, but he cannot be identified with our Savery. It has been stated that he was a military engineer, but there was no such recognized corps in the army at that date, although persons were attached to line regiments for entrenching and such-like duties. Such might have been Savery's status, and some support is lent to this supposed military part of his career by the fact that in 1705 he published a *Treatise on Fortifications*, translated from the Dutch of Baron Coehoorn. A person who had been employed as "trench-master" in the army might well have received the courtesy title of captain on retirement. Another supposition is that Savery was a mine captain in Cornwall, where the title "captain" is given to mine managers, both underground and surface, but there is no evidence that he ever had such employment, while at the same time it is inherently improbable, from what we know about him otherwise. Then again it is suggested that he was a sea captain, but beyond the fact that he invented a ship's log in 1710, and that he wrote a pamphlet entitled *Navigation Improved*, 1698—wherein, however, he disclaims knowledge of maritime affairs—we have nothing upon which to base such a supposition. We should have liked, too, to have known something about Savery's appearance, but the portrait of him which is believed to have once existed is not now to be found.

Savery must have been in favour at Court, for in 1705 he was appointed, by the good offices it is said of Frederick Prince of

Vales, Treasurer to the Sick and Wounded Seamen broken in he Dutch Wars, a post that must have taken up much of his ime, and one that he did not relinquish till 1714, when he was ppointed Surveyor of the Water Works at Hampton Court. Ie was elected a Fellow of the Royal Society in 1705.

What is certain is that Savery was, to judge by the evidence f the Patent Rolls, the most prolific inventor of his day; no ess than seven patents, or applications for such, between 1694 nd 1710, stand in his name. The only one of Savery's patents hat is of importance for the present enquiry, however, is that r a "new Invention for Raiseing of Water and occasioning Motion to all Sorts of Mill Work by the Impellent Force of Fire, which will be of great vse and Advantage for Drayning Mines, erveing Towns with Water, and for the Working of all Sorts f Mills where they have not the Benefitt of Water nor constant Vindes", dated July 25, 1698 (No. 356), and covering England, Vales and Berwick-on-Tweed. The grant was for the usual eriod of fourteen years, but in the following year an Act of 'arliament (10 & 11 William III, cap. 61) extended the period f protection for twenty-one years or a total of thirty-five years n all from the date of the original grant. The Act recites the erms of this grant, and gives this as the reason for its extension:

> And whereas the said Thomas Savery hath by his great care, paines nd expence since the granting the said Letters Patents greatly nproved the said Invention...but may and probably will require nany yeares' time and much greater expence than hitherto hath been o bring the same to full perfection, and the said Thomas Savery hereby deserves a better encouragement by having a longer terme f yeares allowed him for the sole use and benefitt of the said Invention....

nd so on with legal prolixity.

Protection for the invention was extended to Scotland in 1701 y an Act of the Scottish Parliament, granting to James Smith f Whitehill in the county of Midlothian the sole right to use an engyne or invention for raiseing of water and occasioning notion of millwork by the force of fire". Smith's petition set ut what had taken place in England, viz. "that by Articles of

Agreement signed betwixt the said Captain Thomas Savery and the petitioner...he hath allowed and given to the petitioner the communication of the said invention with the model of the said engine to be made use of by him within this Kingdom upon the conditions and provisions contained in the said articles". The protection was to terminate at the same date as that of the extended patent for England.

It is pertinent to enquire how and why Savery turned his attention to steam; all we know on this subject is what he tells us in his own words taken from the dedication of his work: *The Miner's Friend*[1] to the "Gentlemen Adventurers in the Mines of England". It reads thus:

> And though my thoughts have been long imployed about water-works, I should never have pretended to any invention of that kind had I not happily found out this new, but yet a much stronger and cheaper form or cause of motion than any before made use of. But finding this of rarefaction by fire, the consideration of the difficulties the miners and colliers labour under by the frequent disorders, cumbersomness, and in general of water-engines, incouraged me to invent engines to work by this new force, that tho' I was obliged to incounter the oddest and almost insuperable difficulties, I spared neither time, pains, nor money till I had absolutely conquer'd them.

We would give a great deal to know how he came by his knowledge of the "rarefaction by fire", and what were the "almost insuperable difficulties" that he overcame; we should like too, to know what means he had in view of "occasioning motion of millwork"; doubtless it was by pumping water back on to a waterwheel from the tail race of the wheel, as he hints in the pamphlet in question. It is true we have some gossip as to the genesis of Savery's invention, gossip that was current at a very early date (see p. 52). This then is the story as retailed by Dr John Theophilus Desaguliers (1683–1744):[2]

> CAPTAIN *Savery*, having read the Marquis of *Worcester*'s Book, was the first who put in Practice the raising Water by Fire, which he

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[1] *The Miner's Friend, or an Engine to raise Water by Fire described, and the manner of fixing it in Mines, with an account of the several other uses it is applicable unto; and an answer to the objections made against it*, by Tho. Savery, Gent., London, 1702.

[2] *Experimental Philosophy*, II, 1744, p. 465.

proposed for the draining of Mines. His Engine is describ'd in *Harris's* Lexicon (see the Word *Engine*) which being compared with the Marquis of *Worcester*'s Description, will easily appear to have been taken from him; tho' Captain *Savery* denied it, and the better to conceal the Matter, bought up all the Marquis of *Worcester*'s Books that he could purchase in *Pater-Noster-Row*, and elsewhere, and burn'd them in the Presence of the Gentleman his Friend, who told me this. He said that he found out the Power of Steam by Chance, and invented the following Story to persuade People to believe it, *viz*. that having drank a Flask of *Florence* at a Tavern, and thrown the empty Flask upon the Fire, he call'd for a Bason of Water to wash his Hands, and perceiving that the little Wine left in the Flask had filled up the Flask with Steam, he took the Flask by the Neck and plunged the Mouth of it under the Surface of the Water in the Bason, and the Water of the Bason was immediately driven up into the Flask by the Pressure of the Air. Now he never made such an Experiment then, nor designedly afterwards, which I thus prove:

I MADE the Experiment purposely with about half a Glass of Wine left in a Flask, which I laid upon the Fire till it boil'd into Steam: then putting on a thick Glove to prevent the Neck of the Flask from burning me, I plung'd the Mouth of the Flask under the Water that fill'd a Bason; but the pressure of the Atmosphere was so strong, that it beat the Flask out of my hand with Violence, and threw it up to the Cieling. As this must also have happened to Captain *Savery*, if ever he had made the Experiment, he would not have fail'd to have told such a remarkable Incident, which would have embellish'd his Story.

The story of Savery burning all the copies he could find of a book thirty-five years after its publication sounds rather far-fetched. There is no improbability, however, in the story of the wine flask, in spite of Dr Desaguliers having repeated the experiment with different results. The doctor, we regret to say, seems to have been only too ready to detract from the performances of others than himself, and to attribute bad motives to them, as we shall see again later.

Seeing that we have little more than gossip as to the stages gone through in the development of the invention, and though it would illuminate our minds to know the truth of the matter, we must pass on to the practical exemplification of Savery's

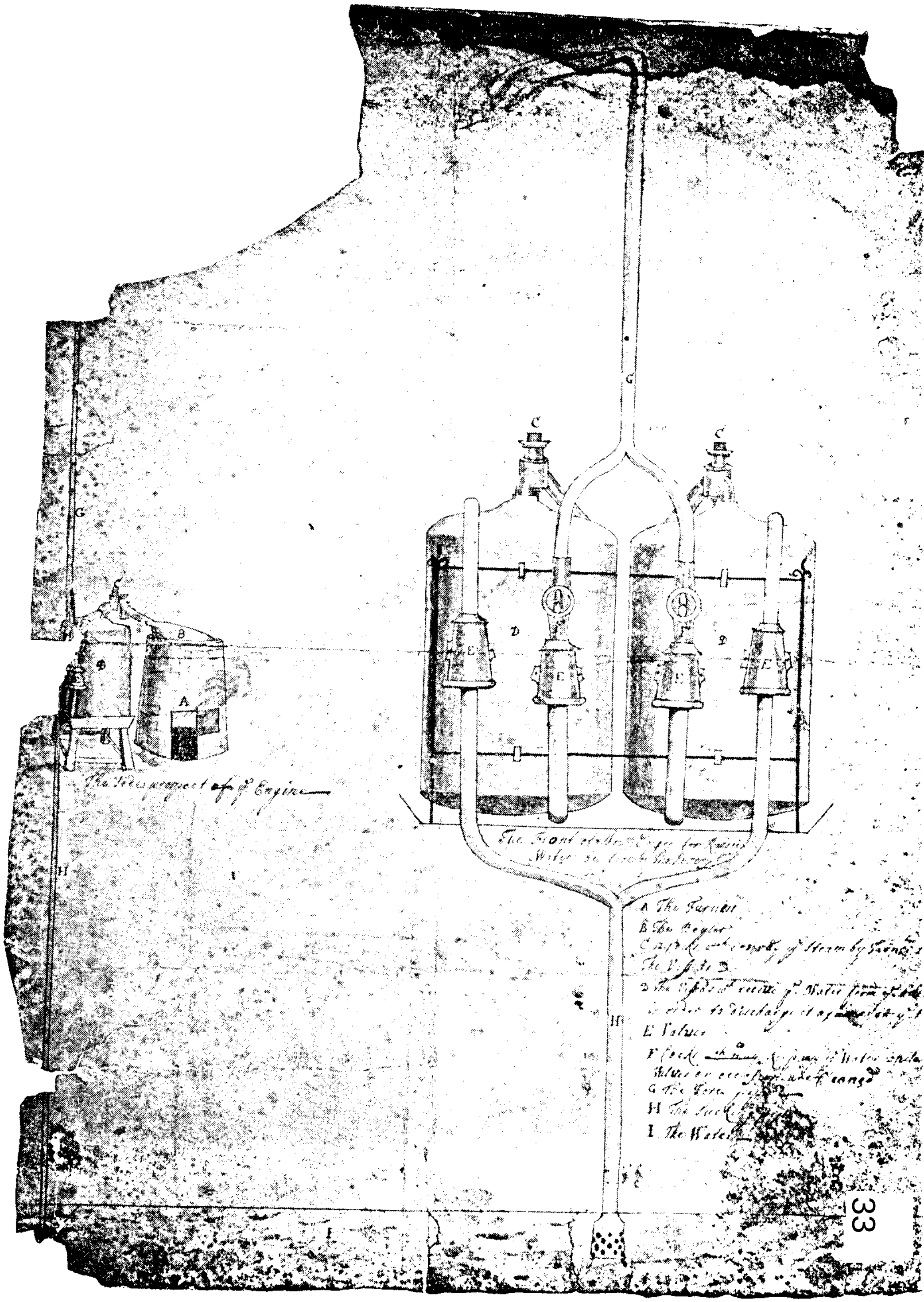

PLATE I. SAVERY'S FIRE ENGINE 1699

ideas in his engine. Patentees, at the time of which we are writing, were not required to file specifications of their inventions, and Savery was no exception to this practice. However, he made a working model of his engine, which he exhibited to the King at Hampton Court Palace—incidentally, this shows that he, as we have said, had access at court; a little later, viz. on June 14, 1699, he showed the model to the Fellows of the Royal Society. The following is the minute of the Meeting: "Mr. Savery Entertained the Royal Society with shewing a small model of his engine for raising water by the help of fire, which he set to work before them; the experiment succeeded according to expectation, and to their satisfaction." This, as Savery says, was "its first appearance in the world". On this memorable occasion, besides demonstrating the model to the Fellows present, Savery furnished the Society with a drawing of his engine; fortunately this has been preserved (see Pl. I). As the legend on it is difficult to read we give the transcript, as follows:

> The Front of An Engin for Raising Water by fire by Tho: Savery.
> The side prospect of ye Engine:
>
> *A.* The Furniss.
> *B.* The Boyler.
> *C.* 2 Cocks wch convey ye Steam by Turns to The Vessells *D.*
> *D.* The Vessells wch receive ye Water from ye bottom in order to discharge it againe att ye top.
> *E.* Valves.
> *F.* Cocks wch may Keep up ye Water while ye Valves on occasion are Cleans'd.
> *G.* The Force pipe.
> *H.* The Sucking pipe.
> *I.* The Water.

It is from this drawing that the engraving in the *Philosophical Transactions*[1] was prepared, where the account of the engine appears.

It will be observed that there are two receivers, one of which is filled by suction while the other is emptied by pressure, and

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[1] *Phil. Trans.* 1699, xxi, No. 253, p. 228.

so on, thereby ensuring a nearly continuous discharge of water. The engine is supported by an ornamental iron frame and stands on a tray on a stool. The boiler seems to be little more than an alembic and no chimney is shown to the fireplace.

In the three years next ensuing, the apparatus underwent considerable improvement in detail, as is shown by the illustration and description in *The Miner's Friend* already referred to, viz. the addition of a hand gear for operating the two steam cocks together; the provision of non-return valves to the receivers which saved perhaps a foot in height; the installation of an auxiliary boiler to replenish the main one and the supplement of a cistern to supply the condensing water.

Savery must have formed very high hopes of the success of his engines, for he set up a workshop for their production, as the following advertisement shows:[1]

> Captain Savery's Engines which raise Water by the force of Fire in any reasonable quantities and to any height being now brought to perfection and ready for publick use, These are to give notice to all Proprietors of Mines and Collieries which are incumbred with Water, that they may be furnished with Engines to drain the same, at his Workhouse in Salisbury Court, London, against the Old Playhouse, where it may be seen working on Wednesdays and Saturdays in every week from 3 to 6 in the afternoon, where they may be satisfied of the performance thereof, with less expense than any other force of Horse or Hands, and less subject to repair.

Salisbury Court extends from Fleet Street towards the river Thames, near St Bride's Church. This was the first workshop established in the world for making steam engines, and the advertisement is the first to announce such manufacture commercially.

The better to understand the principle and mode of action of Savery's engine, it will be preferable to consider its early and simple form rather than the one shown in *The Miner's Friend.* Fortunately, we have such an example described and illustrated by Richard Bradley, F.R.S.[2] (Fig. 6). This engine was erected apparently by Savery himself or his workmen in or about 1712, at Campden House, Kensington, to raise water for domestic purposes. Bradley regarded it as the "truest proportioned of any about London", and it "succeeded so well that there has not been any want of water since it has been built". Referring to the illustration it will be seen that the apparatus comprised a globular boiler *B*, with a steam pipe *D*, in which was a plug stopcock *C*, supplying steam to a bottle-shaped receiver *E*, in one with a flat box *F*. The latter had a pipe *G*, dipping into the well or water supply below where there was a non-return valve. The box had a rising main *L* also with a non-return valve, and it delivered by a launder *O* to a tank or reservoir. Steam was admitted to the receiver and when it was quite hot the stopcock *C* was shut, and water from the cock *M* was allowed to pour over the outside. The steam inside was condensed, and the atmosphere forced the water up the main *G* to fill the receiver. Steam was admitted over the surface of the water which was thus forced up the rising main *L*, where it was again retained by a non-return valve. The cycle could be repeated four times per minute by turning the appropriate cocks; as the receiver held 13 gallons of water, and as the height of the suction pipe was 16 ft. and of the rising main 42 ft., the useful work done by the engine was about 1 h.p. The prime cost of the apparatus was £50.

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[1] *Post Man,* March 19–21, 1702.

[2] *New Improvements of Planting and Gardening,* 2nd ed. 1718, p. 175.

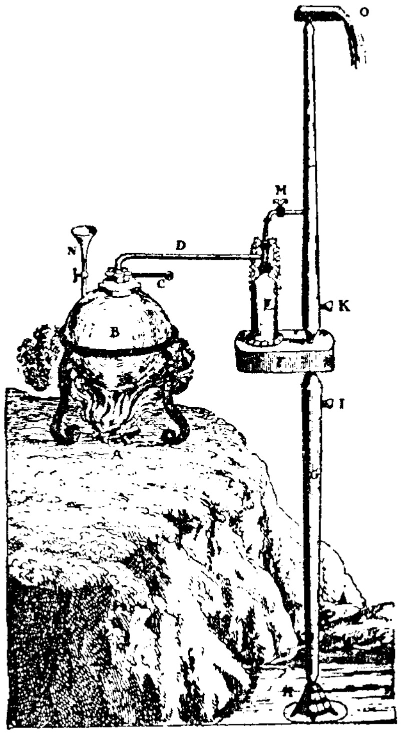

Fig. 6. Savery's fire engine at Kensington, 1712.
From Bradley's *New Improvements of Planting and Gardening*, 1718.

It will be observed that the actual total lift of the engine was

of the order of 50 ft., so that for the purpose for which the engine was primarily intended—that of raising water from mines—there would have to be at intervals of 50 ft. vertically in the pit a series of engines each delivering into the sump of the one above. This meant a boiler and engine, sat at every 8 fathoms of the depth—in fact, such a multiplicity of plant—for mines had already reached, exceptionally, a depth of 50–60 fathoms—that the apparatus was hardly to be thought of; in fact, we do not know with certainty that any plant was ever installed in a mine, even in a shallow one. What the plant was like that he intended to put down in a mine is shown in the illustration taken from *The Miner's Friend*. Bearing in mind the description already given, it will be observed that he had, by this date, made several improvements; the vessels or receivers are in duplicate, such that by acting alternately they gave a greater discharge at a more even rate than did a single receiver. A second boiler is provided to heat the water to supply the second. May we not look upon this as the first boiler feed-water heater? The cocks for controlling the operations are brought within the convenient reach of one man. Nevertheless, there was no essential difference between this apparatus and the one first brought out. We know that one of these engines was installed at York Buildings in the Strand, for supplying the neighbourhood with a domestic supply from the Thames, but even this "was liable to so many disorders if a single mistake happened in the working of it, that at length it was looked upon as a useless piece of work and rejected".[1]

To sum up, the engine found a limited application where the suction lift was about 20 ft. and the forcing lift somewhat of the same order, as in the case of supplying water to a mansion, but for the purpose of raising water from mines, it could not, and did not, fulfil the sanguine hopes held out for it. Savery dropped it incontinently after a few years' experience, possibly at the time when he was appointed Treasurer to the Sick and Wounded, i.e. in 1705.

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[1] Bradley, Richard, *Ten Practical Discourses concerning Earth and Water*, 1727, p. 33.

Desaguliers puts the matter in a nutshell when he says:[1]

> Captain *Savery* made a great many Experiments to bring this Machine to Perfection, and did erect several, which rais'd Water very well for Gentlemen's Seats; but could not succeed for Mines, or supplying Towns, where the Water was to be rais'd very high and in great Quantities: for then the Steam requir'd being boil'd up to such a Strength, as to be ready to tear all the Vessels to pieces. . . . I have known Captain *Savery*, at *York*-Buildings make steam eight or ten times stronger than common air (i.e. 117 to 147 lb. per sq. in.); and then its Heat was so great that it would melt common soft Solder; and its Strength so great as to blow open several of the Joints of his Machine; so that he was forc'd to be at the Pains and Charge to have all his Joints solder'd with Spelter or hard Solder.

If true this would certainly be the case as the temperature of steam of ten atmospheres pressure is 357° F. and soft solder melts below that.

Some reader may remark that it would have been easy to increase the height of lift by increasing the boiler pressure; so it would were it not that the state of the constructional arts of the time was not sufficiently advanced to admit of making boilers to stand the pressure so required. The failure of the engine must be attributed in the main to imperfections of workmanship, and the unreliability of materials. To-day there would be no difficulty in making a successful engine to Savery's drawings, indeed, in a modified form and with an automatic valve, Savery's principle is embodied in that most useful emergency pump, Hall's pulsometer of 1876, still manufactured extensively.

It is curious, by the way, to observe what an attraction Savery's idea has had, for one inventor after another has dabbled in it. We must mention Papin, who turned aside in 1707 from the true line of advance, i.e. the cylinder-and-piston engine, to make an engine on Savery's plan. Papin, who was then at Hesse-Cassel, submitted his proposal to the Royal Society with a request for a grant of money to carry out experiments upon it. The Society naturally referred the proposal for report to Savery, their member most qualified to do so. He found nothing in it that he had not already done himself, and did not recommend the grant. In the course of his criticism, there is a noteworthy passage in which he goes out of his way to damn the cylinder-and-piston engine; he considered it was impossible to make it work, because the friction set up would be too great! This is significant as we shall see when we come to Newcomen.

The only other inventor of the Savery type engine that we need mention is William Blakey, because he went further than the rest and made his design of the engine fully automatic. This was in 1776,[1] but alas he was at the heel of the hunt, for long before this time a safe and fairly efficient engine for raising water had been invented, and to this we must devote the next chapter.

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[1] *Experimental Philosophy*, ii, 1744, p. 466.

[1] *Trans. Newcomen Soc.* xvi, p. 97.

HOERBIGER INFORMATION

CHECK VALVES AND THEIR EFFECT ON PULSATIONS

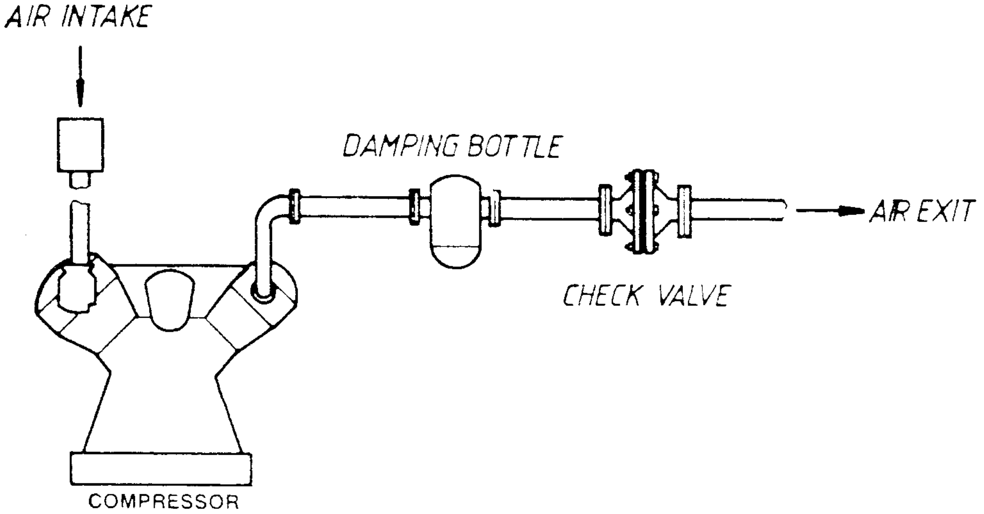

FIG. 1 RECIPROCATING COMPRESSOR WITH DAMPER AND CHECK VALVE

Reciprocating compressors are equipped with automatic suction and discharge valves, which act as check valves; one allowing gas to enter the cylinder and the other to discharge it after compression. In addition, for safety reasons a separate check valve is usually installed in the piping downstream from the compressor. Process considerations, too, may require the downstream installation of further check valves. Typically for this purpose the piping engineer has a choice of several types of check valves that include the flapper-type, swing-type, piston check and the plate-type check valve, which is similar in design to the compressor valve.

A special study was conducted using a Hoerbiger Depend-A-Check[1] valve, to prove that plate-type check valves act as pulsation dampers.

Single-acting piston compressors discharge the compressed gas intermittently or, more exactly, in a cyclic manner during about one-quarter of a crankshaft turn. This cycle creates pressure waves called pulsations. Double-acting compressors discharge gas twice as often, and compressors with several cylinders in parallel, more often still. Although this makes the flow more uniform, pulsations are still present though at a correspondingly higher frequency.

The pressure peaks of a pulsating flow spread out in the gas at the speed of sound and, in addition, are carried away by the flow at its velocity.

Pulsations in industrial plants are undesirable for a number of reasons. They may, for example, impair the process by disturbing the orderly performance of flow sequences, create cyclic stress loading on the materials and vibrations in the piping system. The consequences thereof may be damages to the plant equipment and production problems. Some frequently reported mechanical damages are cracks in piping, heat exchanger or vessel welds; fatigue failures of heat exchanger tubes; damage to pipe supports; and the spontaneous loosening of nuts, bolts and anchor bolts. Vibrations also result in noise emissions, particularly in the case of building vibration.

In general, pressure peaks in pulsating flows decrease gradually due to turbulent flow, mixing and friction as the gas flows down the pipe. These effects are influenced by factors such as flow velocity, gas characteristics, roughness of pipe wall and duct geometry. However, the decrease is so gradual that most industrial plants require additional measures to subdue the pulsations. In contrast, pressure peaks may also increase. This occurs when the pipe geometry is of such a shape that the system is in resonance (the systematic vibration of gas in the pipe — similar to a pipe organ). The gas vibration under these conditions may considerably increase the pressure waves discharged by the compressor resulting in highly destructive forces.

To soften the pulsation, a damping bottle is usually installed in the discharge piping. A damping bottle is a vessel equipped with internal baffles, such as perforated plates, that help to even out the pressure peaks by acting as orifices that cause turbulent gas mixing.

Compressor valves are designed for operation in pulsating flows, and so are many plate-type check valves. Fig. 2 and Fig. 3 show the details of a type of compressor valve frequently used. When in the closed position the valve plate covers the seat. The plate in the case under consideration is fixed by a flexible arm, the elastic force of which is boosted by additional spring plates. Hence, the valve closes against the flow direction with declining pressure load. With the valve plate in lifted or open position, the gas passes through a number of ring shaped ports in the seat, plates and guard. The designer has the option to vary several parameters, such as the number of ports, the port width and the lift height, so that the flow and geometry are tuned to the operating conditions.

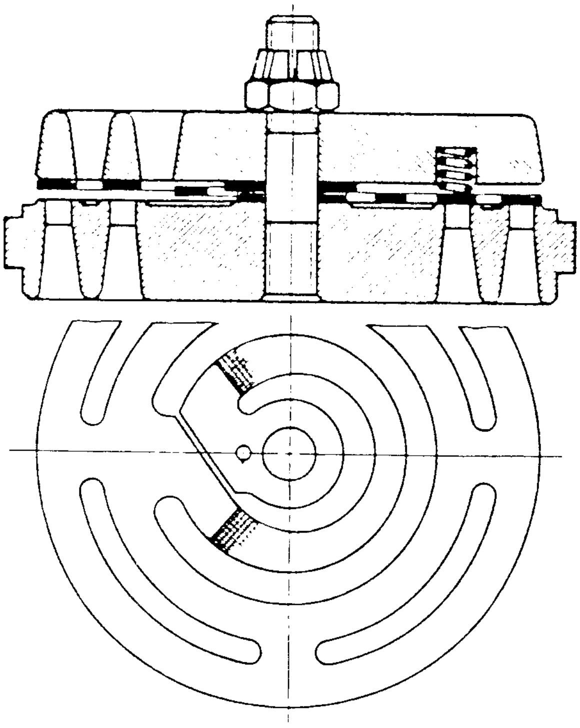

FIG. 2 FRICTIONLESS GUIDED PLATE VALVE WITH FLEXIBLE GUIDE ARMS (Hoerbiger®)

The ring shaped ports in the various valve parts form strip-type passage areas that act as orifices followed by abrupt widening of the cross-section resulting in eddy formation, as in the case of the perforated plate. Also, the various gas streams flow towards each other, thus effecting a further evening out of the pressure peaks. Therefore, it may be expected that a valve of this design will exert a damping influence on a pulsating gas flow, as simulated graphically in Fig. 4.

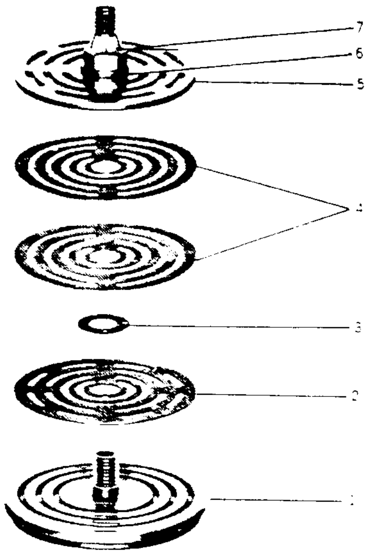

FIG. 3 EXPLODED VIEW OF A FRICTIONLESS GUIDED PLATE VALVE (Hoerbiger®)

1. **Valve with center bolt assembled and dowel pinned**
2. **Valve plate**
3. **Lift washer, middle**
4. **Spring plate**
5. **Guard**
6. **Washer, if required**
7. **Lock nut**

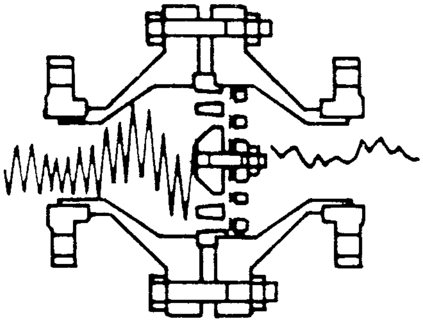

FIG. 4 GRAPHICAL REPRESENTATION OF EXPECTED DAMPING CAUSED BY A PLATE-TYPE CHECK VALVE

To prove this, a number of tests under conditions similar to field environments were conducted in Hoerbiger's test facility in Vienna, Austria. The test set up was basically as shown in Fig. 1, but without a damping bottle. A one-stage reciprocating compressor was used to discharge air into a piping system in which a check valve of the type shown in Fig. 3 was installed. To study various design parameters the valve could be removed and changed. The aim was to measure the cyclic pressure variations at various points of the piping. Pressure transducers installed at these points produced signals that were fed to a calibrated oscilloscope. The analysis of the gathered test data proved that a considerable damping of the pulsation was achieved.

Fig. 5 shows on top the oscillogram produced by a pressure transducer installed just upstream of the check valve, and beneath it one generated by a transducer just downstream from it. The curves represent the pressure variations versus time. The pressure peaks caused by the discharging of the compressor can be clearly seen to spread across about a quarter of a cycle. For reasons of clarity the zero point of the lower curve was shifted slightly below that of the upper curve. The pressure peaks were measured and yielded the result that the pulsation intensity downstream from the check valve had decreased by as much as 25%.

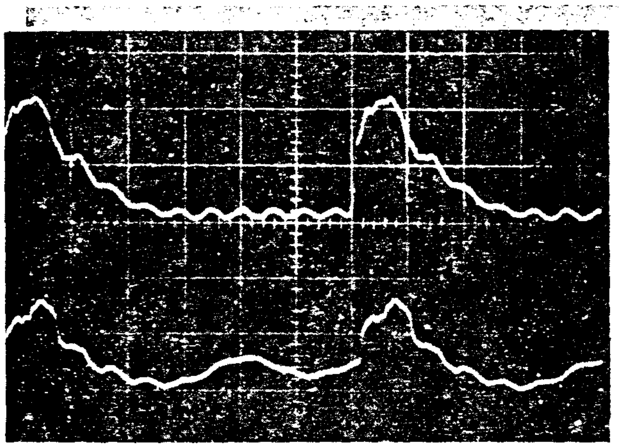

FIG. 5 OSCILLOGRAM OF PULSATIONS UPSTREAM TO AND DOWNSTREAM FROM A PLATE-TYPE CHECK VALVE

Fig. 6 shows further test results. In this test the lift of the valve plate was reduced by approximately one-third as compared to the previous test. The oscillogram shows a decrease of the pressure amplitude by as much as 32%.

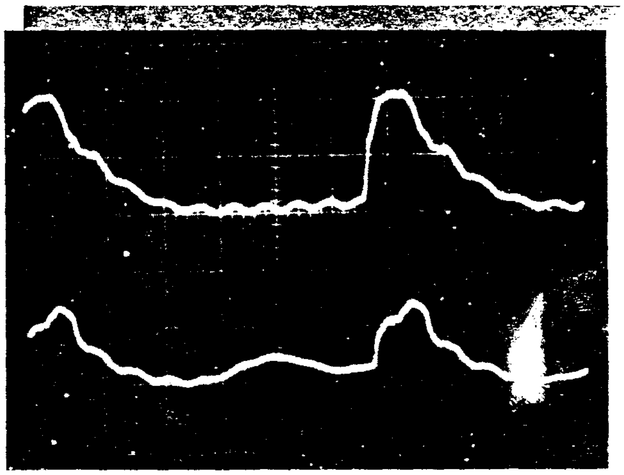

FIG. 6 OSCILLOGRAM OF PULSATIONS UPSTREAM TO AND DOWNSTREAM FROM A PLATE-TYPE CHECK VALVE WITH REDUCED LIFT

During the first test the ratio of the free flow passage area in the valve to the free area in the pipe section had been 0.53; in the later case it was 0.36. It had been expected that the damping effect would grow with decreasing area ratio based on the following simple thought: The bigger the volume of the upstream buffer vessel — in this case, the piping — and the smaller the free passage area of the nozzle — in this case, the check valve — the more uniform will be the downstream flow if a pulsating flow is fed to the system. This principle was verified in the test.

To avoid the above-mentioned gas vibrations, the system has to be "out of tune", so that the exciting pulsation does not meet with resonance. Frequently, however, for process, design or space reasons there is no way to modify the shape of the flow duct, or even to shift its supports. For example, the pipeline may be supported by the structure of a building, the frames of which obviously cannot be moved. In such a case, by installing a plate-type check valve a considerable softening of the vibration can be achieved, as was proved in other simulation tests. By inserting this element into a suitable spot in the flow system, the latter may be de-tuned.

SUMMARY

For various reasons, a check valve is usually installed in the piping downstream from a reciprocating compressor. This type of compressor discharges a pulsating flow, which has to be smoothed to avoid undesirable influences on the process and damage to the plant equipment.

By means of testing it was proved that insertion of a plate-type check valve (Hoerbiger Depend-A-Check®) caused damping of the pulsation by as much as 32%. Thus, a plate-type check valve in the discharge piping serves two purposes:

It acts as a check valve and as a pulsation damper.

The physical phenomena involved are of a complex nature and rather difficult to express in mathematical formulas. The test results found may not be directly applied to other cases. There is, however, the qualitative statement that damping definitely is effected. Its intensity will have to be determined for each application separately.

In the case of strong flow pulsations or when a particularly uniform flow downstream from a reciprocating compressor is required, it may be appropriate to install a damping bottle plus a plate-type check valve in series.

HOERBIGER

Mailing Address: P. O. Box 8888
Ft. Lauderdale, Florida 33310

Shipping Address: 1381 S.W. 30th Avenue
Pompano Beach, Florida 33069

Telephone: 305/974-5700 or Toll Free 800/327-8961
Florida WATS 800/334-6902
FAX 305/974-0964

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Fluid Transients in Fluid Structure Interaction
p. 93-99

CHECK VALVE BEHAVIOUR UNDER TRANSIENT FLOW CONDITIONS
A STATE-OF-THE-ART REVIEW

A.R.D. Thorley
Thermo-Fluids Engineering Research Centre
The City University
London, United Kingdom

ABSTRACT

Recent advances in the understanding of check valve behaviour are reviewed. It is evident that the basis now exists for providing reliable information on valve dynamics that is in a form that can be easily assimilated and used by pipeline design engineers. The use of non-dimensional valve performance curves is discussed and recommended. Manufacturers are encouraged to provide this data. Design engineers are charged with the responsibility of using it and are provided with guidelines for so doing.

INTRODUCTION

Check valves are items of equipment that were originally developed and fitted to pipelines in order to prevent the lines draining backwards when pumps stopped - and sometimes also to prevent the downstream reservoirs from emptying. In addition they prevented reverse rotation of pumps, thereby avoiding damage to seals and to the brush gear of the driving motors.

It was thought, initially, that the valves simply had to close fairly quickly and to seal effectively. The early designs, therefore, were based in general on the concept of a simple swinging disc (closure element). These were usually quite adequate to protect relatively low head systems but as pumping pressures increased, multi-pump systems evolved and surge protection devices such as air vessels became more common, so too did the phenomenon of "check valve slam".

In these simple swing check valves, in which the moving element has to travel through a significant distance, e.g. 60-90°, and in which its position is controlled almost entirely by the movement of the fluid, the mean velocity of the fluid is backward at the instant of closure. This reverse flow velocity is reduced immediately to zero giving rise to high pressures and the valve slam phenomenon. The name is derived from the fact that as the valve nears its seat, reducing the backward flow, a pressure rise builds up which accelerates the closure, so that at the last moment the valve disk does indeed "slam" onto its seat.

This problem has assumed increasing importance in recent years and led to various test programs, modifications to existing valves, and the development of new styles of valve. For several years some degree of confusion existed over the most appropriate remedies for the problem of valve slam and the criteria that should be applied when selecting the most suitable check valve for a particular application.

Through the efforts of a number of researchers (1-12) the behaviour of check valves is now understood more clearly, enabling them to be selected with greater confidence in their performance.

SYSTEMS MOST AT RISK

Piping systems and check valves that are most at risk are those where pumps are delivering against relatively high pressure heads. In particular one can cite examples of multi-pump installations such as deep mine drainage, power station condensate and boiler feed pumps where major problems are known to have occurred.

For example, Figure 1 shows a section of a cooling water system comprising four pumps in parallel and which has experienced major problems with check valve slam. Normally up to three pumps are in service with one as a standby. The pumps are rated at 50 m head when delivering 833 ℓ/s at a speed of 980 rev/min and having a power requirement of 470 kW.

As a result of attempting to protect the pump with a carelessly selected swing check valve major problems of check valve slam, line vibration and failure occurred. Figure 2 illustrates pressure-time histories obtained from controlled tests following a pump trip when, in the upper trace, a swing check valve was fitted. For comparison purposes, the lower trace is for the same conditions but with a nozzle type check valve used instead.

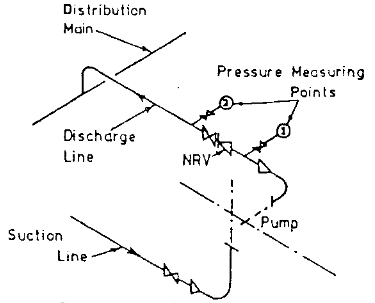

Fig.1. Layout of one of four similar feed pumps arranged in parallel showing the location of the valves and pressure measuring points.

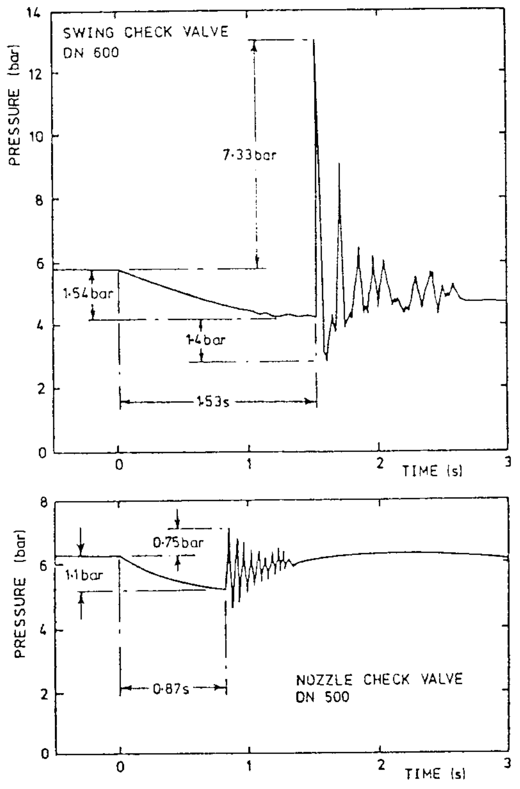

Fig.2. Comparison of pressure-time histories immediately downstream of the check valve (point 2) in Figure 1. Three pumps were in service initially; this one was tripped out at zero time.

The test conditions were that three pumps were running and one, for which the results are given, was tripped out. The only differences between the two sets of tests were (a) the nominal diameters of the swing and nozzle check valves were 600 mm and 500 mm respectively and (b) the pump impeller was reduced from 650 mm to 620 mm diameter. However, the initial steady conditions were of the same order for both tests and yet the results are markedly different.

Another example is illustrated with Figures 3 and 4 which refer to a power station condensate extraction system. Several comparisons have been made (13) but the figures shown concern a manual pump changeover. The sequence in both cases is:

Pump 'B' running - Start Pump 'A'
- Trip out Pump 'B'.

Figure 3 shows the pressure-time response when the pumps are protected by swing check valves; Figure 4 is the corresponding response when nozzle type non-return valves were fitted instead.

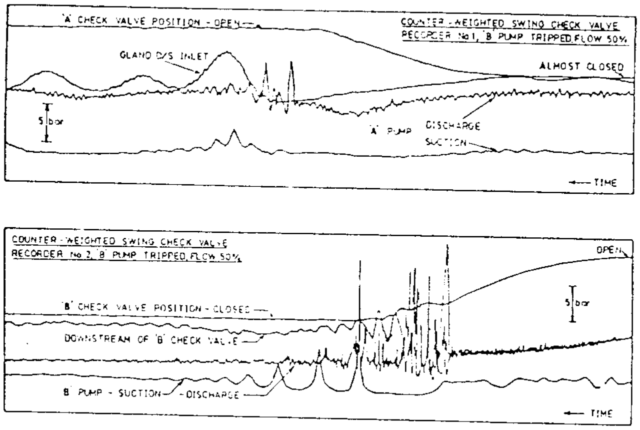

Fig.3. Pressure-time records for a manual pump changeover when the pumps are fitted with counter-weighted swing check valves.

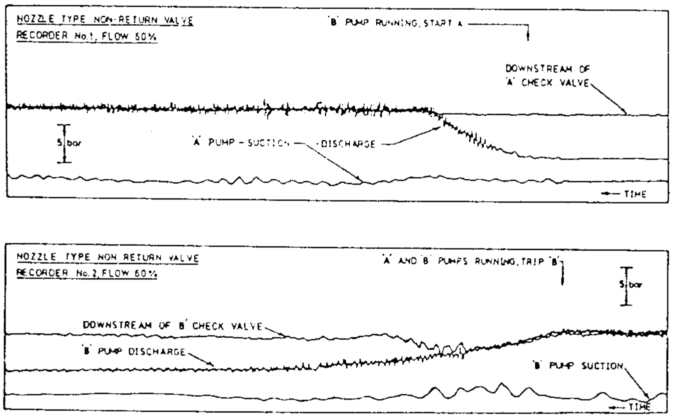

Fig.4. Figure 3, but with the pump fitted with nozzle type check valves.

These are two of several multi-pump examples (14) that could be quoted by way of illustration. However, even single pump installations can be at risk, particularly if they supply a pipeline protected by an air vessel. In such a case, when the power driving the pump is lost the air vessel acts as an energy source analogous to other pumps continuing to run in parallel. Liquid is forced out of the air vesel and into the system tending to maintain forward flow down the pipeline but also tending to flow back through the pump. If the check valve is slow in responding ideal conditions for hazardous check valve slam are created. Figure 5 is an example pressure-time history for a laboratory test on such a system.

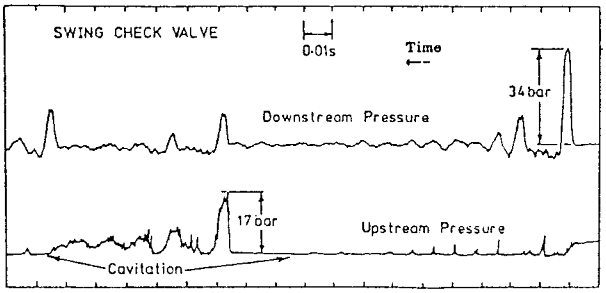

Fig.5. An example of upstream and downstream pressure-time histories for tests on a swing check valve in a single line following a simulated pump trip Note that 'time' goes from right to le

Since the final flow stoppage is virtually instantaneous, the pressure transient created on the donwstream side of the valve is a 'rapid' event in the context of fluid transients and the magnitude ΔH of the pressure head rise caused by the reduction in flow velocity from V_R to zero may be estimated from the well-known Joukowski equation, i.e.

$$\Delta H = \frac{cV_R}{g} \qquad (1)$$

where c is the speed of propagation of transient pressure waves in the pipeline (typically 1000-1200 m/s in steel pipelines) and g is acceleration due to gravity. In approximate terms:

$$\Delta H \simeq 100\ V_R \qquad \text{metres head.}$$

It is therefore imperative that the maximum reverse velocity V_R, which occurs virtually at the instant of closure, should be reduced as near to zero as possible. This can only be achieved by careful design and selection of check valves which must be matched to the system of which they are an integral part.

It should be noted also that the occurrence of high transient pressures is not restricted to the downstream side. On the upstream side of the closing valve, as the reverse velocity is reduced to zero, the falling pressure may easily drop to the vapour pressure of the liquid and a vaporous cavity produced. This cavity will normally collapse rapidly in response to pressure waves reflected from further upstream giving rise to large pressure pulses which may, in some circumstances, cause the check valve to re-open momentarily and then crash back onto its seat. The remedy is to select a valve that is capable of responding sufficiently quickly to the changing flow conditions so that pressure fluctuations due to the valve closure itself are within acceptable limits. This does not necessarily mean that closure must occur at the instant the flow velocity is zero, the pipeline may be capable of withstanding a rapid shut-off from a small forward or reverse velocity.

Equally it is misleading to define or specify valve responses in terms of a required closure time since it is not this in itself that is important, but the flow velocity at the instant of closure. This is dependent on the system being considered and although a particular time delay is acceptable for a given valve on one system it would usually be quite inappropriate for another.

CHARACTERISING THE DYNAMIC BEHAVIOUR OF CHECK VALVES

In seeking to understand how check valves respond to rapidly changing flow conditions engineers have had two principal motives:

(i) to establish the desirable characteristics of check valves to provide a basis for improved design and selection, and

(ii) to develop suitable mathematical means of describing valve response for inclusion in computer codes used to predict fluid transient behaviour of pipelines.

These objectives prompted two lines of research, a variety of experimental studies (1,2, 4,8, 11-14) in which several valve types were studied independently and under different circumstances, and theoretical studies (3,5,6,10, 11,12,14,15) which in the main referred to attempts at developing equations of motion for the valves, although two (12,16) introduce the use of dimensional analysis and dynamic similarity to enable experimental data to be expressed in terms of useful equations.

DESIRABLE CHARACTERISTICS

The empirical studies previously referenced were undertaken on both prototype plant and test facilities in engineering laboratories. Despite the results having been produced from a variety of sources there were sufficient comparisons that some meaningful conclusions could be drawn (5,14). The principal ones were that a good response to changing flow conditions could be obtained if:

(i) the moving parts of the valve are of low inertia,
(ii) their motion is assisted by a spring or springs, and
(iii) the distance/angle through which the moving elements have to travel is small.

The extent to which actual valves possess these features can be illustrated graphically in the form of Dynamic Performance Characteristics after the fashion first proposed by Provoost (4) - see Figure 6.

The vertical axis is the maximum reverse velocity that occurs during valve closure, usually immediately before the moving elements are seated. The base line is the mean deceleration of the liquid column as the flow is brought to rest. In practice, this is the slope of the main part of the velocity-time trace, which seems to be fairly constant in many cases.

On such a performance chart, the ideal curve would be coincident with the horizontal axis. The swing check and ball valves which are the subject of the upper curves are clearly a poor match, whereas the nozzle valve with a strong spring is the best suited to the system.

To select a check valve for a particular system the procedure to be adopted is to determine the deceleration of the liquid column (dV/dt) under expected transient or fault conditions and then by reference to valve performance characteristics select a valve that has an acceptable dynamic response - i.e. not necessarily the best, but one that is adequate for the job. In some systems swing check valves may still be good enough, but in others a very high level of performance may be essential.

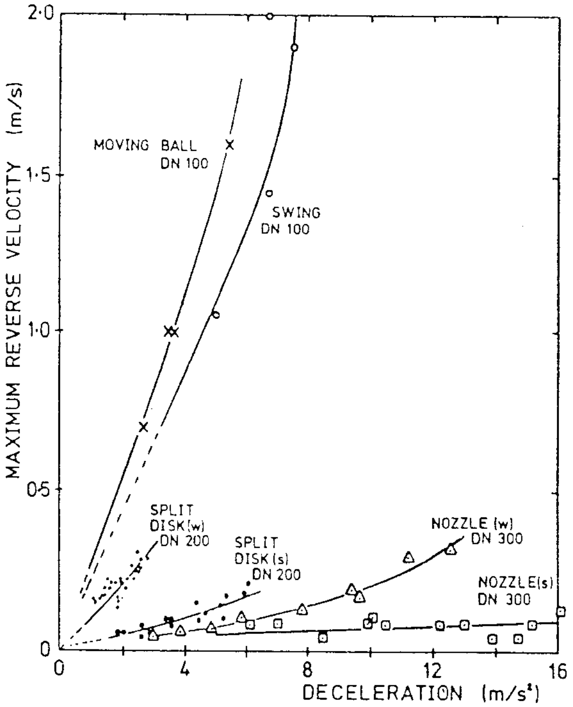

Fig.6. Typical dynamic performance characteristic, after the style of Provoost (2,3) which should be provided by valve manufacturers.

Before proceeding to a discussion of how these experimentally determined dynamic characteristics can be used in conjunction with computer codes, the purely theoretical developments which were evolving concurrently will be mentioned briefly, for completeness.

THEORETICAL STUDIES

When a valve, or any other component, is incorporated into a mathematical model of the unsteady flow through a piping system it is necessary to have equations relating the pressure change 'ΔH' across it to the flow 'Q' through it, i.e.

$$\Delta H = f(Q) \qquad (2)$$

The precise form of the equation depends on the nature of the component, and for valves can include experimentally determined loss coefficients and the state of opening or closure as preferred. For check valves this leads into a need for equations relating the forces and torques to the kinematic behaviour of the moving elements. For example, for swing check valves, torques due to gravity, T_G, fluid pressure,T_P, friction T_f, buoyancy T_B, and springs T_S, are related to the angular acceleration $\ddot{\theta}$ by:

$$T_G + T_P + T_f + T_B + T_S = (I_D + I_F)\ddot{\theta} \qquad (3)$$

The right hand side of this equation contains the inertia I_D and I_F of the valve door and adjacent fluid respectively.

When the valve elements move axially, as in nozzle and plunger type valves (3) the general form of the corresponding equation of motion is similar to equation (3). Forces replace torques and masses replace inertias.

The gravitational, buoyancy and externally applied torques, together with the inertia of the moving elements in the valve, are easily determined from valve dimensions and material properties. The remaining terms require some degree of speculation supported by carefully controlled experiments for the deduction of empirical coefficients. For example, it may be postulated that the friction torque, T_f, would be of the form:

$$T_f = K_1 + K_2\,\theta^n \qquad (4)$$

in which K_1, K_2 are constants and n an arbitrary, and perhaps constant, power.

Similarly, hydrodynamic forces/torques will depend on the rapidly changing flow patterns around the valve, its position and the flowrate Q, e.g.

$$T_P = f(Q,\theta,\dot{\theta},\ddot{\theta}) \qquad (5)$$

The precise form cannot be found analytically - a probable form might be supposed and empirical coefficients determined for particular examples.

The remaining inertia term, I_F, refers to the added mass effect of the fluid moving in close proximity to the valve door. It is somewhat speculative, but from experiments in air and water (17) it does appear that the added mass effect for a circular swinging gate valve could be estimated as being equivalent to a sphere of fluid having a diameter the same as the valve door and with their geometric centres coincident.

Due to the indeterminate nature of some of the components of equation (3), and the wide variety of valve types and sizes, recent developments in the form of non-dimensional groups (14,12) especially when allied (12) to the Provoost style dynamic characteristics would appear to show most promise for inclusion in computer models of transient flow.

SCALING LAWS AND DIMENSIONLESS CHARACTERISTICS

The benefits of model testing in thermo-fluids engineering are well-known and such techniques are frequently used - pumps, turbines, aerofoil sections etc., being common examples. In particular, model testing frequently helps keep development costs within acceptable limits as well as providing important insights into fluid flow phenomena.

An essential pre-requisite of model testing is that geometric similarity must exist between model and prototype. This helps ensure kinematic similarity, i.e. similarity of flow pattern, which in turn leads to dynamic similarity or similarity of forces and torques.

The hope is then that data collected from experiments with models can be expressed in terms of suitable non-dimensional parameters which will apply to both the model and the prototype.

Combining and extending ideas proposed by Collier and Hoerner (16) and Koetzier et al (12) parameters that are relevant to the dynamic behaviour of many non-return valves are:

D	characteristic diameter
dV/dt	mean fluid deceleration
V_o	flow velocity when the valve is just fully open (critical velocity)
V_R	maximum reverse velocity just prior to closure
ρ_m	density of the moving elements
ρ_f	density of the fluid
F_o	net spring and gravitational force on the moving element when just fully open
F_s	net spring and gravitational force on the moving element when closed
μ	fluid viscosity
Δp	frictional pressure drop across the valve.
x	distance travelled by the moving element from its seat.

These eleven parameters, which between them contain the three fundamental properties of mass, length and time, can be arranged into eight independent groups using one of the several techniques that are available (18), e.g. Buckingham-Pi theorem or the Rayleigh method.

Taking ρ_f, D, and μ as the repeating variables the following groups can readily be developed:

$$\frac{D}{V_o^2} \cdot \frac{dV}{dt} ; \frac{V_R}{V_o} ; \frac{F_o}{\rho_f V_o^2 D^2} ; \frac{\Delta p}{\rho_f V_o^2} ; \rho_f \frac{V_o}{\mu} D ; \frac{\rho_m}{\rho_f} ; \frac{F_o}{F_s} ; \frac{x}{D}$$

This set really applies to valves in which the moving element translates in an axial direction. A similar set can, however, be derived for swing type valves and split disc types, in which case the forces would be replaced by torque, T_o/linear characteristic dimension. The third and last parameters would then be:

$$\frac{T_o}{\rho_f V_o^2 D} \quad \text{and} \quad T_o/T_s$$

Complete matching of these groups between models and prototypes will be impossible, but some can be ignored as being of minor importance and the others categorized into two groups: (a) those of primary importance and (b) those of secondary importance which should be kept as nearly consistent as reasonably possible.

Of minor importance will be the non-dimensional critical force, $F_o/\rho_f V_o{}^2 D^2$, frictional pressure drops and viscous effects generally in terms of Reynolds numbers. Collier and Hoerner (16), for example, showed from a series of tests that at various opening positions of a split disk (dual plate) valve the ratio $\Delta p/\rho_f V_o^2$ was essentially constant.

It will be noted that the first two non-dimensional groups contain the principal terms of the dynamic characteristic shown in Figure 6. This points to the idea that dimensionless characteristics can be expressed in equation form as:

$$\frac{V_R}{V_o} = \Phi\left(\frac{D}{V_o^2} \cdot \frac{dV}{dt} , \frac{\rho_m}{\rho_f} , \frac{F_o}{F_s} , \frac{x}{D}\right) \qquad (6)$$

and also suggests ways to develop experimental programs to provide the maximum amount of useful information from the minimum number of tests. For example, Figure 7 (taken from Koetzier et al (12)) is a plot of V_R/V_o versus $D/V_o^2.dV/dt$, assuming all other three parameters in equation (6) are constant. Figure 8 is a similar diagram for a nozzle type of non-return valve. Three sets of data are shown, having been obtained from tests into the effect of spring strength on valve response. It is observed that the normal and weak springs give essentially the same response although there is a slight deviation with the strong springs. Given that mean fluid decelerations are used to obtain these data points the degree of scatter for preliminary results of this nature is not unreasonable and indicates that this approach has definite promise.

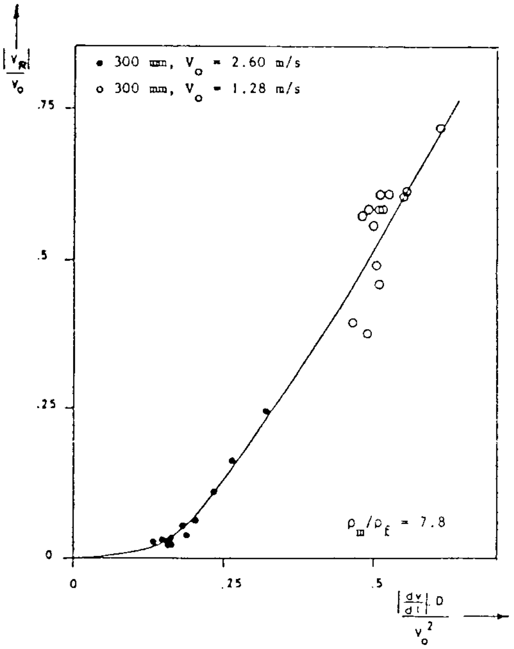

Fig.7. Dimensionless dynamic characteristic for a non-return valve.

All of the results to which reference has been made relate to tests in water. From equation (6) one of the dimensionless parameters is ρ_m/ρ_f, the density ratios of the moving element and the fluid in the pipeline. No attempt has yet been made to assess how closely it is necessary to conform to this ratio when other fluids are being pumped. For example, if, say, oil of relative density 0.875 is being pumped, how important is it that the valve elements be manufactured from a material correspondingly lighter to obtain the same dynamic response? Future tests with different fluids and valve elements of different materials will provide an indication.

REPRESENTATION OF CHECK VALVE BEHAVIOUR IN COMPUTER CODES

The most common assumption in computer codes is to assume that the valves close at the instant the flow reaches zero at that location, even though it may be recognised that this is not quite true. Some attempts have been made to incorporate a delay using techniques such as swept volume during closure, e.g. based on reverse velocity multiplied by the area of the moving element, but it has been acknowledged (ref.11, for example) that these methods can also be quite inaccurate.

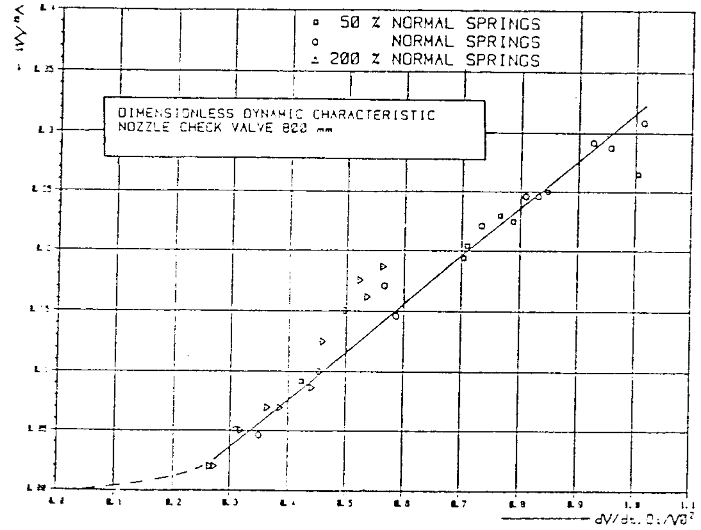

Fig.8. Dimensionless dynamic characteristic for a 800 mm nominal bore nozzle type check valve for three spring strengths.

If, as a result of incorporating data from several valve sizes in the same homologous groups, it transpires that a single curve results, a much better and simple representation of check valve behaviour will be possible. A strategy for achieving this is as follows:

(i) From a knowledge of the permissible maximum pressure in the pipeline and valve casing (due to acceptable stress and shock load levels) specify an acceptable pressure head change ΔH due to a sudden stoppage.

(ii) From equation (1), obtain the maximum permitted reverse velocity V_R, by solving:

$$V_R = g\Delta H/c \qquad (1)$$

(iii) Normalise V_R by dividing by V_o, the flow velocity when the valve is just fully open - sometimes taken from experience or by design as 75% of the intended flow velocity. This will give an upper limit on charts such as Figure 7.

(iv) In the computer code to be used for transient analysis, the boundary condition representing the check valve should arrange for the valve to close at the instant the flow velocity reaches the value of V_R.

(v) From the predicted velocity-time history at the valve up to this point the mean fluid deceleration is obtained. Normalising this with v_o and D, and reading up on Figure 9 to intersect with the corresponding V_R/V_o line enables a valve with a suitable dynamic response to be selected.

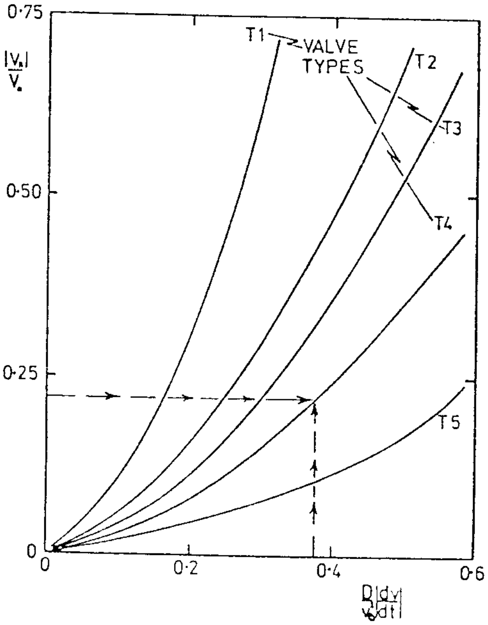

Fig.9. Indication of how non-dimensional valve performance curves may be used for check valve selection. Valves T4 and T5 would be suitable.

(vi) The pressure-time histories should also be scrutinised carefully to confirm that the maximum pressures experienced adjacent to the valve do indeed remain within acceptable limits.

In order to adopt the foregoing strategy it is essential that adequate dynamic response curves are available. It has been argued previously (9,12) that just as pump and turbine manufacturers are expected to produce performance curves for their equipment, so too should manufacturers of check valves. It is even more true now than before.

RESPONSIBILITIES AND LIABILITIES

The pipeline design engineer responsible for specifying components for his systems has a duty to ensure that they are the most appropriate and cost-effective. In the context of check valves, as with other components, he can only do this if he is provided with adequate data. For example, he must have all the information necessary to follow through the procedures outlined in the previous section - which in turn means that:

(i) Valve manufacturers must accept the responsibility for providing authentic performance data on their products. Eventually, insurance companies may insist upon it.

(ii) The designers responsible for the overall system should ensure that fluid transient studies are carried out for the proposed system, encompassing all the possible scheduled and unscheduled operating scenarios that are likely to produce hazardous conditions - and the 'worst case' adopted as the design criteria.

In recognition that many systems are not installed as designed or operated as intended it is also the responsibility of the engineer in overall charge to review the situation at the commissioning stage. It is only then that certain assumptions such as wave propagation speeds, pump performance data, friction losses, etc., used in the fluid transient studies, can be confirmed.

CONCLUDING REMARKS

The last 6-10 years has witnessed notable advances in the understanding of check valve behaviour. A basis has been established for graphically representing valve dynamic response from measurements taken under laboratory conditions. Non-dimensional representation of the important terms maximises the use to which such data can be put and provides useful guidelines for future valve testing.

The non-dimensional performance curves also enable check valves to be represented by simple boundary condition routines in computer programs. The key feature is to establish the maximum acceptable reverse velocity through the valve and use this as the control parameter in the program. The mean deceleration of the fluid then provides the vital link to selecting the most appropriate check valve commensurate with meeting cost and performance criteria.

ACKNOWLEDGEMENTS

Figure 8 is included by permission of the Delft Hydraulics Laboratory and the manufacturers who sponsored the work.

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THE CHATTERING OF SAFETY VALVES AND SIMILAR MECHANISMS

534.1 : 621.646.4

It often happens with safety valves that rapid oscillations (chattering) occur which are very disturbing. In the following it will be shown that the elasticity of the fluid (steam, gas, oil foam) is the cause of these oscillations and that the oscillating parts gain energy from the fluid. An oscillation, once it has begun, always increases in the absence of a minimum amount of damping. The question is mathematically examined and this minimum amount of damping indicated.

SAFETY valves and similar mechanisms include a piston which is acted upon on one side by the fluid pressure to be regulated and on the other by a spring. When the fluid pressure increases, the piston uncovers a larger exit opening and in this way regulates the pressure. A mechanism of this type is schematically shown in Fig. 1. To examine the question of chattering, it is assumed that the piston oscillates about a middle position x_m, the stroke of the piston being indicated by x. If the fluid is incompressible and the pump discharge constant then the pressure is always given by the instantaneous position of the piston and is, of course, least for the upper position of the piston and greatest for the lower position (the change in the volume V due to the movement of the piston being neglected). The stroke of the piston is plotted against time at the top of Fig. 2 and the pressure p underneath. While the piston is moving downwards from B to C, the pressure varies from b—c and averages p_m. While the piston is moving upwards from C to D the pressure again averages p_m. The energy transmitted by the fluid to the piston during its upward movement is equal to the energy transferred to the fluid by the piston during its downward movement. This means that the piston in moving neither gains nor loses energy and when damping is present (friction of the piston in the guide-way) the oscillation must gradually die away.

Now, however, the alteration of the volume V due to the movement of the piston should be taken into consideration. It is easy to appreciate that this has a damping effect. If, for example, more fluid flows in, the valve

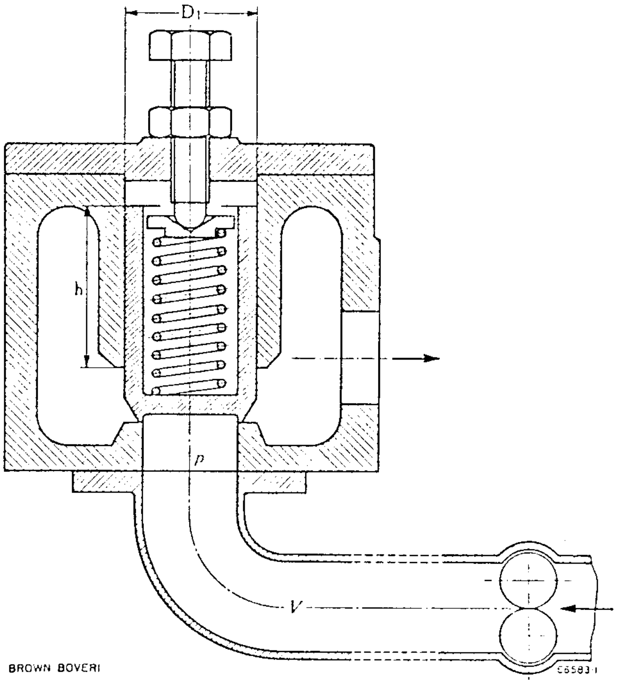

Fig. 1. — Section through a normal spring-loaded safety valve

V = Volume of the piping between pump and valve
p = Pressure of the fluid
h = Length of the piston guide-way

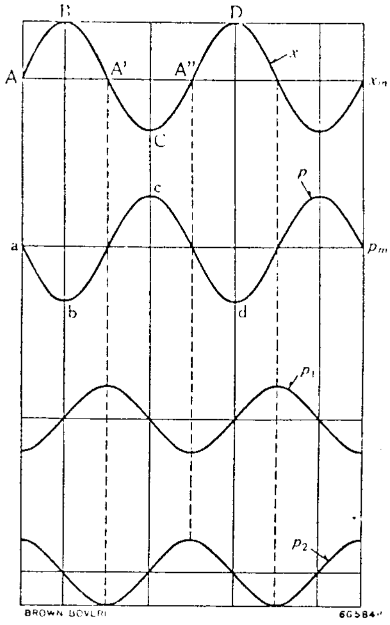

Fig. 2. — Variation of the pressure as a function of time during oscillation of the piston

Top curve = Stroke of the piston as a function of time
Second curve = Pressure of the fluid under the piston
p_1 = Additional pressure caused by the change in volume due to the movement of the piston
p_2 = Additional pressure due to the elasticity of the fluid

must open still further; this movement of the piston provides extra space which can accommodate some of the fluid. Consequently the valve need open somewhat less than would be necessary without this increase in volume. Moreover, the pressure need not increase so much. On the other hand, if less fluid flows in, the piston forces fluid into the space V and the pressure decreases less. These alterations of the pressure can be considered as an additional pressure change which adds to the pressure p. This additional pressure change depends on the velocity of the piston. The greater the velocity of the piston, the greater is this additional pressure change. This is shown as p_1 in Fig. 2. The total pressure then becomes $p + p_1$. Since, as has been shown, the piston in moving neither gains nor loses energy, only the effect of p_1 need be considered. It is now clear that while the piston is moving downwards from B—C, the pressure p_1 is greater than while the piston is moving upwards from C—D. The energy taken from the fluid by the piston as it moves upwards is less than the energy transmitted to the fluid by the piston as it moves downwards and *this constitutes a damping effect because energy is taken from the piston.* This damping effect supplements the previous damping (friction of the piston in the guide-way). With an incompressible fluid therefore any oscillation present should quickly die away.

The process is easy to understand mathematically.

Let

F_K = Sectional area of the piston on which the fluid pressure acts, in cm^2.

F = Instantaneous opening which the valve uncovers for the exit of the fluid.

x = Instantaneous position of the valve.

p = Instantaneous fluid pressure.

Q_0 = Constant quantity of fluid in cm^3/s supplied by the pump.

Then

$$x = x_m + \Delta x = x_m\left(1 + \frac{\Delta x}{x_m}\right) = x_m\,(1 + \xi)$$

$$p = p_m + \Delta p = p_m\left(1 + \frac{\Delta p}{p_m}\right) = p_m\,(1 + \pi)$$

$$F = F_m + \Delta F = F_m\left(1 + \frac{\Delta F}{F_m}\right) = F_m\,(1 + \xi)$$

Now the equation of continuity is

$$Q_0 = \varphi\,F_m\,(1 + \xi)\,\sqrt{p_m}\,(1 + \pi) + F_K\,\frac{dx}{dt}$$

where φ represents the exit flow coefficient. It will be seen that some of the fluid entering flows away and some fills the extra space provided by the piston.

The square root can be expanded, giving:

$$Q_0 = \varphi\,F_m\,(1 + \xi)\,\sqrt{p_m}\left(1 + \frac{\pi}{2}\right) + F_K\,x_m\,\xi'$$

If the valve is motionless in the middle position, then

$$Q_0 = \varphi\,F_m\,\sqrt{p_m}$$

and further

$$F_K\,x_m\,\xi' + Q_0\,\xi + \frac{Q_0}{2}\,\pi = 0 \quad \ldots . \; (1)$$

Further the equation of movement for the valve body is

$$m\,\frac{d^2x}{dt^2} = F_K\,(p_m + \Delta p) - (k\,x + P_v) - c\,\frac{dx}{dt}$$

Here m represents the mass of the piston, k the spring constant, P_v the initial tension of the spring, i.e. the force of the spring when $x = 0$ and c the damping constant.

$$m\,x_m\,\xi'' + c\,x_m\,\xi' + k\,x_m + k\,x_m\,\xi + P_v = $$
$$= F_K\,p_m + F_K\,p_m\,\pi$$

but

$$k\,x_m + P_v = F_K\,p_m$$

therefore

$$m\,\xi'' + c\,\xi' + k\,\xi = \frac{F_K\,p_m}{x_m}\,\pi \quad \ldots . \; (2)$$

Further, introduce

$$\delta_p = \frac{p_{\max} - p_{\min}}{p_m} \quad \ldots\ldots . \; (3)$$

It can be quite well assumed that, in the middle position, the pressure amounts to

$$p_m = p_0 + \frac{p_{max} - p_{min}}{2} = p_0 + \frac{1}{2}\,p_m\,\delta_p$$

where p_0 represents the pressure for the position $x = 0$. Then

$$F_K\,p_0 = P_v$$

and

$$F_K\left[p_0 + \frac{1}{2}\,p_m\,\delta_p\right] = P_v + k\,x_m$$

from which

$$\frac{F_K\,p_m}{x_m} = \frac{2\,k}{\delta_p}$$

Inserting this into (2), then

$$m\,\xi'' + c\,\xi' + k\,\xi = \frac{2\,k}{\delta_p}\,\pi \quad \ldots . \; (4)$$

Adding equation (4) to equation (1) it follows after a simple calculation that

$$m\,\xi'' + \left[c + \frac{2\,F_K^2\,p_m}{Q_0}\right]\xi' + k\left(1 + \frac{4}{\delta_p}\right)\xi = 0 \quad (5)$$

This is the equation for a damped oscillation. The value $k\left(1+\frac{4}{\delta_p}\right) = \approx 25\,\mathrm{k}$ replaces the spring constant k. The alteration of the volume due to the movement of the piston has therefore the same effect as if the spring were considerably stiffened and this explains why the oscillations are so rapid. The damping factor consists of two parts, the first being represented by the friction and the second by the additional damping produced by the alteration of the volume due to the movement of the piston. It is interesting to compare these two factors.

Assume $D_1 = 4$ cm, $h = 6$ cm, the radial play $s = {}^1/_{100}$ mm $= {}^1/_{1000}$ cm. The friction force then is

$$R = \mu F \frac{w}{s} = \mu \cdot \pi \cdot 4 \cdot 6 \cdot 1000 \frac{dx}{dt}$$

from which the damping constant c

$$c = 75\,\mu\,10^3$$

For oil at 50°

$$\mu = 1{\cdot}5 \times 10^{-7}\ \frac{\mathrm{kg\,s}}{\mathrm{cm}^2}$$

giving

$$c \approx \frac{1}{100}\ \frac{\mathrm{kg\,s}}{\mathrm{cm}}$$

Further assume $F_K = 7{\cdot}75$ cm^2, $p_m = 5$ kg/cm^2, $Q_0 = 3$ l/s $= 3000$ cm^3/s, from which the second factor becomes

$$\frac{2\,F_K\,p_m}{Q_0} = 0{\cdot}2\ \frac{\mathrm{kg\,s}}{\mathrm{cm}}$$

It can be seen, therefore, that *the additional damping is about 20 times greater* than the damping due to friction, even though the play chosen is very small (1/100 mm).

However, it is a very different matter when the *fluid is elastic.* For a first estimation the alteration in volume due to the movement of the piston is left out of consideration. While with an incompressible fluid the pressure p follows the movement of the piston, with an elastic fluid the storage capacity must be considered in addition. During the time A A′ the valve is open more than the average x_m implies and during the time A′ A″ closed more. During the time A A′ more fluid flows out than the average stroke implies and during the time A′ A″ less. At A′ the pressure is least and at A″ greatest. To the pressure p an additional pressure p_2 must be added, which is greatest at A and A″ and least at A′. This additional pressure p_2 is also shown in Fig. 2. Further, we need only consider the supply of energy due to this pressure. While the piston is moving downwards from B to C, the average pressure is less than p_m and while the piston is moving upwards from C to D, it is greater than p_m, i.e. the energy which the fluid imparts to the piston on its upward path is greater than the energy transmitted to the fluid by the piston on its downward path. This means that energy is imparted to the moving piston and an oscillation when once begun must continue to increase. If this is to be avoided, then more energy must be dissipated by the two previously mentioned damping effects than is produced.

Again, this process is easy to understand mathematically. The continuity equation is

$$F_K\,x_m\,\xi' + Q_0\,\xi + \frac{Q_0}{2}\,\pi + Q_k = 0$$

Here Q_k represents the extra quantity which can flow in due to the compressibility of the fluid.

Assume that the fluid at the pressure p_m be fluid oil foam consisting of βV cm^3 of air at pressure p_m and of $(1-\beta)V$ cm^3 of oil. In time dt, $\beta Q_k\,dt$ cm^3 of air and $(1-\beta)\,Q_k\,dt$ cm^3 of oil are introduced. At the disposal of the air there thus remains a volume

$$V^* = \beta V - (1-\beta)\,Q_k\,dt$$

In this the following weight must be present

$$G^* = \beta V \gamma_m + \beta\,Q_k\,\gamma_m\,dt$$

Now as weight = volume × specific weight, it follows that

$$\beta\,\gamma_m V + \beta\,Q_k\,\gamma_m\,dt = [\beta V - (1-\beta)\,Q_k\,dt]\,[\gamma_m + d\gamma]$$

from which

$$Q_k = \frac{\beta V}{\gamma_m}\,\frac{d\gamma}{dt}$$

Now the relation between γ and p must be considered. It is assumed that the air is compressed *isothermically.*

Then

$$p\,v = \text{const.}$$

further

$$\gamma = c_1\,p$$

and

$$\frac{d\gamma}{dt} = c_1\,\frac{dp}{dt}$$

Hence

$$Q_k = \frac{\beta V}{p_m}\,\frac{dp}{dt} = \beta V \pi'$$

Consequently the equation of continuity is

$$\beta V \pi' + \frac{Q_0}{2} \pi + F_K x_m \xi' + Q_0 \xi = 0 \quad . \; . \; (6)$$

The equation (4) for the movement of the piston remains unchanged and from (4) and (6) it follows after a simple calculation that

$$m \xi''' + \left[c + \frac{m Q_0}{2 \beta V}\right] \xi'' + \left[k + \frac{F_K^2 p_m}{\beta V} + \frac{c Q_0}{2 \beta V}\right] \xi' + \\ + \left(1 + \frac{4}{\delta_p}\right) \frac{k Q_0}{2 \beta V} \xi = 0 \; . \; . \; . \; . \; . \; (7)$$

This is the equation of an oscillation. If the oscillation is to be damped, the following stabilizing condition must be fulfilled:

$$\left[c + \frac{m Q_0}{2 \beta V}\right] \left[k + \frac{F_K^2 p_m}{\beta V} + \frac{c Q_0}{2 \beta V}\right] > m \left(1 + \frac{4}{\delta_p}\right) \frac{k Q_0}{2 \beta V}$$

or

$$c^2 + \left[\frac{2 \beta V k}{Q_0} + \frac{m Q_0}{2 \beta V} + \frac{2 F_K^2 p_m}{Q_0}\right] c > \frac{4}{\delta_p} k m \quad \frac{m F_K^2 p_m}{\beta V} \; . \; . \; . \; . \; . \; . \; . \; (8)$$

This can be more closely investigated by an example. Assume $V = 3000$ cm³, which includes the volume of the oil in the valve and also the volume of the oil in the oil pipe as far as the pump. Further assume

$$k = 20 \text{ kg/cm}, \quad Q_0 = 3000 \text{ cm}^3/\text{s}, \quad m = 2 \cdot 10^{-4} \frac{\text{kg s}^2}{\text{cm}}$$
$$F_K = 7{\cdot}75 \text{ cm}^2, \quad p_m = 5 \text{ kg/cm}^2, \quad \delta_p = 0{\cdot}155.$$

The value δ_p is calculated as follows: At 5 kg/cm² pressure the velocity of the oil flowing out is 33·3 m/s. The diameter of the valve is 3·1 cm; assuming an exit flow coefficient $\varphi = 0{\cdot}62$, it follows that $x_m = 0{\cdot}15$ cm and since $F_K p_m \delta_p = 2 k x_m$ then $\delta_p = 0{\cdot}155$.

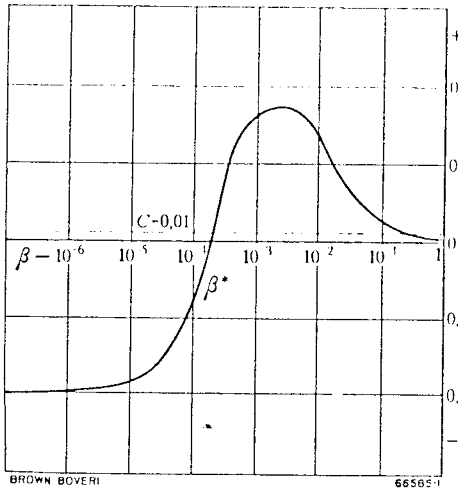

Fig. 3. — Necessary damping constants as a function of the quantity of the air in the oil, to ensure stability

From this equation (8) becomes

$$c^2 + \left(40 \beta + \frac{1}{10^4 \beta} + 0{\cdot}2\right) c > 0{\cdot}102 - \frac{1}{50\,000 \beta} \quad (9)$$

If the sign = is inserted instead of the sign > then the minimum value of c as a function of β results. This value is shown in Fig. 3. A logarithmic scale has been chosen for β. The following is now apparent: For $\beta = 1$, i.e. pure air, $c \geqq 0{\cdot}0026$. The value $c = 0{\cdot}01$, which represents the maximum damping effect attainable, with a piston as in Fig. 1 (play = 0·01 mm), is likewise inserted. Thus β must be $> 0{\cdot}2$, i.e. quite abnormal quantities of air must be present in the oil to ensure stability of the valve. An oil mixed with air in this way is much too elastic and thus cannot be used for regulating purposes, for instance. The other possibility is that $\beta < 0{\cdot}0002$, representing extraordinarily small quantities of air which would be considerably exceeded in practice.

To effect the damping of such a mechanism it is necessary to provide either a much stronger damping effect c or a much stronger additional damping effect, so that the point of intersection β^* of c with the β-axis moves towards the right. According to equation (8) the point of intersection lies at

$$\frac{4 k}{\delta_p} = \frac{F_K^2 p_m}{\beta^* V}$$

i.e.

$$\beta^* = \frac{F_K^2 p_m \delta_p}{4 k V} = \frac{F_K x_m}{2 V}$$

If β^* is to increase, then either the volume V (of valve + piping) must be reduced or $F_K x_m$ increased. A reduction of the volume entails smaller pipe sections and greater oil velocities, which would have further drawbacks. Thus $F_K x_m$ must be increased. To what extent this is possible with the valve under consideration will now be investigated.

Assume that the elastic modulus of the oil, which has air mixed with it, is $E = 500$ kg/cm², a value which has been determined experimentally. Since $E \approx p_m/\beta$, then $\beta \approx 0{\cdot}01$ when $p_m = 5$ kg/cm². Instead of $\beta = 0{\cdot}000194$ it must be equal to 0·01, i.e. about 50 times greater. Such an increase is out of the question because, as can easily be seen, the spring would have quite impossible dimensions. Thus the only remaining solution is to ensure a much greater damping effect c. This can be effected by employing a special construction with additional damping, so that all valves or similar mechanisms can be designed to operate without chattering.

(MS 846) *J. v. Freudenreich (D. O'S.)*

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THE DYNAMIC BEHAVIOUR OF NON-RETURN VALVES

G.A. Provoost

Delft Hydraulics Laboratory, The Netherlands.

Non-return valves serve to prevent the reversal of the flow. If, however, the reversal of the flow occurs in a very short time the valve may close when the flow already has been reversed. Depending on the type of non-return valve a sudden decrease of the reversal flow will occur, possibly resulting in unallowable pressure variations and slamming of the valve.

To theoretically predict whether slamming of a valve is to be expected, data under dynamic conditions of this valve have to be known. Since in most cases these data are not known the behaviour of a non-return valve during transient flow may be predicted by means of dynamic tests for several conditions. These tests can be carried out in a circuit which allows a sudden reversal of the flow. A description is given of such a test-rig. With a non-return valve installed in this test-rig, flow rate and upstream and downstream pressures can be measured during decreasing and possibly reversed flow under various conditions.

The results of tests carried out for two types of non-return valves are shown. The results of the measurements are explained and discussed. Tests have shown that especially the fluid velocity gradient with time is of decisive importance when considering the dynamic behaviour of non-return valves.

NOMENCLATURE

a	torque coefficient
c	velocity of propagation of pressure waves
D	pipe diameter
f	friction coefficient according to White-Colebrook
g	acceleration due to gravity
H	pressure head
ΔH	pressure head loss
I	mass moment of inertia
Q	flow rate
t	time-coordinate
T	total torque exerted on valve body
T_e	torque exerted on valve body due to external forces
T_f	torque exerted on valve body due to friction
T_h	torque exerted on valve body due to hydrodynamic pressures
T_w	torque exerted on valve body due to weight of valve body
v	fluid velocity
x	coordinate along axis of pipeline
α	angle of rotation of valve body
ξ	pressure head loss coefficient

1 INTRODUCTION

A non-return valve is a valve which serves to prevent reverse flow. Non-return valves are often used in combination with pumps. When these pumps are switched off the flow through these pumps, and consequently the flow through the non-return valves, will decelerate to zero. If the non-return valves are not closed at the instant of zero flow, the flow will reverse and the non-return valve will close during reverse flow, possibly resulting in unexpected high pressure variations and slamming of the valve.

It is important to know the dynamic behaviour of non-return valves so that in the design-stage of a hydraulic project it is possible to predict whether the chosen type of non-return valve will satisfy. For the purpose a theoretical description of the behaviour of non-return valves during transient flow was set up. Since up to now insufficient data are available only dynamic tests under several conditions may lead to acceptable predictions. The aim of this study was to find the determinants on which the dynamic behaviour of non-return valves depends.

2 THEORY

A non-return valve consists of a valve body, whose position is influenced by the flow (closed, partly open or open position). Generally the valve body will be in the closed position if a pressure drop exists in the direction in which the flow has to be prevented. In this situation no reverse flow occurs. If, however, the direction of the flow is changing from the permitted one into reverse direction in a very short time, it is possible that the valve body is still in a partly open position while reverse flow occurs. Due to the reverse flow the valve body will move fast into the closed position, thus causing a sudden zero flow condition. Two phenomena can be observed at the instant of closure of the non-return valve, viz.

(1) a sudden stop of the movement of the valve body
(2) a sudden stop of the reversed flow through the non-return valve.

The first phenomenon can be noted by a more or less heavy sound depending on the velocity of the valve body at the instant of closure. This phenomenon is often referred to as the slamming of the valve.

The second phenomenon causes a pressure rise at the initial downstream side of the valve. At the same time a pressure drop occurs at the initial upstream side of the valve. This pressure drop may be so large that the vapour pressure of the fluid is reached. In that case a vapour bubble will grow at the initial upstream side of the valve. The disappearance of such a vapour bubble causes a sudden pressure rise which may lead to a re-opening of the valve. The second closure of the valve may again cause reaching of the vapour pressure thus leading to a repeating phenomenon.

To avoid slamming of the valve it may be required that the valve-body reaches its closed position at the instant the flow becomes zero. Whether this occurs may be predicted by comparing the calculated time interval in which the flow in the relevant pipe system decreases to zero and some minimum possible closing time of the valve. Such a method was proposed by Lewinsky-Kesslitz (Ref. 1) who has also given theoretical minimum possible closing times for rectangular swing type non-return valves. However, the use of this method will not always result in acceptable predictions since the motion of the water during the closing procedure is neglected.

Only a more detailed analysis of the motion of the valve body will lead to a good prediction whether slamming will occur or not. Hereto all forces acting on the valve body have to be taken into account. The following consideration is made for valves with rotating valve bodies but will be equivalent for valves with translating valve bodies. The motion of the valve body will depend on the flow rate through the valve and the upstream and downstream pressures. The relationship between these variables is given in the equations given below:

- The motion of a rotating valve body is described by Newton's Second Law of Motion:

$$T = I \frac{d^2\alpha}{dt^2} . \tag{1}$$

where: T = torque of the valve body,
I = mass moment of inertia of rotating parts and a part of the surrounding water, and
α = angle of rotation of the valve body.

The effect of inertia of the fluid surrounding the valve body will be appreciable, especially for fast motion of the valve body, and has to be added to the mass moment of inertia of the moving parts of the valve. As a simplification it has been assumed in equation (1) that added mass is independent of the motion of the valve body and flow around the valve body.

The torque exerted on the valve body is due o the weight of the valve body (T_w), external forces (T_e), friction (T_f) and hyd dynamic pressures (T_h):

$$T = T_w + T_e + T_f + T_h . \tag{2}$$

The torque caused by the weight of the valve body (gravity) depends on the dimensions and geometry of the valve. The torque due to external forces may be caused by a ballast weight or a spring and depends also on the geometry of the valve. The torque caused by hydrodynamic pressures does not depend on the geometry of the valve but rather on the flow around the valve body. Since the motion of the valve body disturbs the steady state flow around the valve body the flow pattern around the valve body will depend on the flow passing the valve, on the position of the valve body, and on the angular speed of the valve body:

$$T_h = f(Q, \alpha, \frac{d\alpha}{dt}) . \tag{3}$$

- The pressure loss created by the valve also depends on the flow pattern around the valve body. Written in terms of pressure head we obtain:

$$\Delta H = g(Q, \alpha, \frac{d\alpha}{dt}) . \tag{4}$$

- The equations describing unsteady one-dimensional pipe flow in both adjacent pipes are obtained from the consideration of the mass conservation of a control-volume for a pipe segment (5) and from the momentum equation along the axis of the flow (6) as given by Bergeron (Ref. 2):

$$\frac{\partial H}{\partial t} + v \frac{\partial H}{\partial x} + \frac{c^2}{g} \frac{\partial v}{\partial x} = 0 . \tag{5}$$

$$\frac{\partial v}{\partial t} + v \frac{\partial v}{\partial x} + g \frac{\partial H}{\partial x} = - \frac{f}{2D} v|v| . \tag{6}$$

These equations hold as long as the actual pressures are higher than the vapour pressure of the liquid. If, however, the actual pressure equals the vapour pressure further expansion of the liquid only leads to the appearance and increase of vapour bubbles, a phenomenon known as cavitation of the liquid. The equations describing this kind of two-phase flow are described elsewhere (Ref. 3).

With equations (1) to (6) and the boundary conditions of the adjacent pipe ends (such as a pump or an air vessel) the motion of the valve body can be predicted.

However, up to know the relationships (3) and (4) are not known. Only by means of dynamic tests the relationships for a certain type of valve can be determined.

This is the reason why the behaviour of a non-return valve under dynamic conditions cannot be predicted accurately.

Only for slow motion of the valve body a prediction of the behaviour of the valve can be made. If the influence of the motion of the valve body on the flow pattern around the valve body may be neglected both the torque exerted on the valve body caused by hydrodynamic pressures and the pressure loss created by the valve will depend only on the flow passing the valve and the position of the valve body.

With the assumption that the flow pattern is independent of the flow rate (no Reynolds effects) the torque and pressure head loss will be proportional to the flow rate squared:

$$T_h = a(\alpha)\; Q|Q|. \qquad (7)$$

$$\Delta H = \xi(\alpha)\; Q|Q|. \qquad (8)$$

The coefficients a and ξ which depend on the position of the valve body have different values for the two possible directions of the flow. These coefficients can be determined experimentally by means of steady state flow tests or theoretically estimated with the assumption of potential flow.

3 DESCRIPTION OF THE TEST-RIG FOR NON-RETURN VALVES

To determine the dynamic behaviour of a certain type of non-return valve the flow through this valve has to be reversed in a very short time. For this purpose a special test-rig for non-return valves was set up in which the initial flow can be reversed in a controlled time interval (see Fig. 1). This test-rig consists of a by-pass on the test-rig for valves of the hydraulic machinery laboratory which forms part of the facilities of the Delft Hydraulics Laboratory. This test-rig for valves is a closed loop circuit equiped with pumps and a pressure control tank and has been described in detail by Oldenziel and Teijema (Ref. 4). The pressure level in the pipe system is adjusted by the pressure control tank in the test-rig for valves. During the tests a certain flow rate is pumped through the test-rig for valves. The flow rate through the non-return valve to be tested in the initially steady state condition can be adjusted by the control valve in the test-rig for valves. The initial upstream side of the test-section of the test-rig for non-return valves is provided with two combined fast acting valves. With this kind of three-way valve the initial upstream side of the test-section can be disconnected from the test-rig for valves and connected with a small air vessel. The pressure in this vessel can be set at a predetermined low value.

With the aid of the combined fast acting valves the pressure at the initial upstream side of the test-section can be lowered in a controlled time to the predetermined value present in the small air vessel. This causes a sudden pressure difference over the test-section leading to a fast, almost linearly decreasing flow and possibly to a reversal of the flow until the non-return valve closes.

The time-dependent flow rate in the test-rig is measured with a fast acting flow-meter. This flow-meter is situated at the initial upstream side of the valve to be tested. The pressures at both sides of the valve are measured with piezo-electric transducers. Both flow rate and pressures are recorded on a UV-recorder.

4 EXPERIMENTS

Experiments were carried out for two types of valves having a diameter of 100 mm each:

(a) a moving ball type non-return valve (see Fig. 2). In this case the valve body consists of a ball which is pushed from its seat by the flow into a slot under normal flow condition.

(b) a swing type non-return valve (see Fig. 3). In this case the valve body consists of a rotating steel plate. At the axis an arm provided with a ballast-weight was installed.

The experiments were carried out for steady state velocities of 1 m/s, 2 m/s and 3 m/s. For each velocity tests were carried out with different pressures in the small air vessel resulting in different gradients for the decrease of velocity in the test section. Moreover for the swing type non-return valve a test was carried out for a smaller distance between ballast-weight and centre of rotation. This test is referred to as experiment No. 7. Time histories of velocity, initially upstream and downstream pressures were recorded for each test. For the swing type non-return valve also the position of the valve body was recorded.

A summary of the experiments shown in this paper is given in table 1.

Experiment No.	Type of non-return valve	Initial steady state velocity	Created pressure drop at initial upstream side
1	moving ball type	1 m/s	15 kPa
2		3 m/s	30 kPa
3		3 m/s	40 kPa
4		3 m/s	50 kPa
5	swing type	1 m/s	60 kPa
6		2 m/s	70 kPa
7		2 m/s	70 kPa
8		3 m/s	75 kPa

Table 1: Summary of experiments.

5 RESULTS

In Figs. 4 to 11 the results of the experiments are shown. These results show that, in each experiment, reversal of the flow occurs and that the valve closes in reverse flow. In table 2 a summary is made of the time interval needed for reversal of the flow, the measured fluid velocity gradients and velocities of the reversed flow at the instant of closure. The positive direction of the flow is taken in the allowed direction of the flow through the non-return valve.

Experiment No.	Time interval for reversal of flow	Fluid velocity gradient	Fluid velocity at closure
1	0,28 s	- 3,6 m/s^2	- 1,0 m/s
2	1,15 s	- 2,6 m/s^2	- 0,7 m/s
3	0,88 s	- 3,4 m/s^2	- 1,0 m/s
4	0,56 s	- 5,4 m/s^2	- 1,6 m/s
5	0,20 s	- 5,0 m/s^2	- 1,05 m/s
6	0,30 s	- 6,7 m/s^2	- 1,45 m/s
7	0,30 s	- 6,7 m/s^2	- 2,0 m/s
8	0,40 s	- 7,5 m/s^2	- 1,9 m/s

Table 2: Summary of measured fluid velocities.

The experiments show that, with the described test-rig, the initial flow can be reversed in a controlled time interval with a nearly constant gradient with time.

In all experiments the valve closed in reversed flow. At the instant of closure a pressure rise occurred at the initial downstream side of the valve and a pressure drop occurred at the initial upstream side of the valve.

From experiments Nos. 2, 3 and 4 (see Figs. 5 to 7) it follows that, for a given steady state flow rate, a larger fluid velocity gradient - which gives a shorter time interval needed for reversal of the flow - leads to larger negative velocities at the instant the valve body (the ball) reaches its seat. And thus with a larger fluid velocity gradients larger pressure fluctuations occur. In experiments Nos. 2 and 3 (see Figs. 5 and 6) vapour pressure is not reached at the initial upstream side of the valve. However, in experiment No. 4 (see Fig. 7) vapour pressure is reached at the initial upstream side of the valve leading to the formation of a concentrated cavity (column-separation). The disappearance of this cavity causes a sudden pressure rise leading to re-opening of the valve and to a repetitive slamming of the valve. Also in experiments Nos. 2 and 3 the valve re-opens after the first closure but no repetitive slamming occurred. The phenomena after closure of the valve in experiments Nos. 1 and 3 (see Figs. 4 and 6) are comparable with each other although the initial fluid velocities were different. The fluid velocity gradients, however, were of the same order. This implies that the fluid velocity gradient dominates the dynamic phenomena during valve closure and that the time interval needed for reversal of the flow is of minor importance.

In all experiments for the swing type non-return valve (see Figs. 8 to 11) the valve body began to move round about the instant of reversal of the flow. This means that friction plays an important role. The valve is in more open position for larger flow rates. In experiment No. 8 (fluid velocity 3 m/s) the valve was in complete open position. A comparison of experiments Nos. 6 and 7 shows that the position of the ballast-weight influences the valve-opening. This means that the position of the valve body during steady state flow depends on the steady state flow rate and the position of the ballast-weight. As the valve body began to move round about the instant of reversal of flow the dynamic phenomena will only depend on the fluid velocity gradient and the position of the valve body during steady state flow. This implies that experiments for steady state fluid velocities larger than 3 m/s will give the same results as an experiment for 3 m/s and the same fluid velocity gradient, since for 3 m/s the valve is already in a fully open position. This is the reason why for large steady state fluid velocities and consequently for large steady state flow rates, the fluid velocity gradient is of decisive importance. For velocities smaller than 3 m/s both fluid velocity gradient and position of the valve body during steady state flow are important. A comparison of experiments Nos. 6 and 7 (see Figs. 9 and 10) shows the influence of the position of the valve body during steady state flow. In both experiments the steady state velocities were 2 m/s and the fluid velocity gradients were - 6,7 m/s. Due to a smaller distance between ballast weight and centre of rotation the valve in experiment No. 7 was in more open position than in experiment No. 6. This implies that in experiment No. 7 the distance the valve body had to travel to closed position was larger than in experiment No. 6. As a result, in experiment No. 7 the time interval between reversal of flow and closure of the valve was larger than in case 6 which is the reason why the absolute value of the fluid velocity at the instant of closure in experiment No. 7 was larger than in experiment No. 6 (velocities were respectively - 2,0 m/s and - 1,45 m/s). For a given geometry of a valve the position of the valve body during steady state flow is only influenced by the steady state flow rate. This is the reason why besides the fluid velocity gradient the steady state flow rate is important when choosing several conditions for dynamic tests of non-return valves.

6 CONCLUSIONS

When theoretically predicting the behaviour of a certain non-return valve, data of this valve under dynamic conditions must be available. Since in most cases these data are not known the behaviour of the non-return valve during transient flow may be predicted by means of dynamic tests for several conditions. When choosing these conditions both steady state flow rate and fluid velocity gradient with time have to be considered. Especially for large flow rates the fluid velocity gradient is of decisive importance. The time interval needed for reversal of flow seems to be of minor importance. The dynamic tests can be performed with the test-rig for valves as described in this paper.

Experiments carried out for two types of non-return valves show that in many practical situations these non-return valves will close in reverse flow. At the instant of closure pressure variations will be initiated. At the initial downstream side of the valve a pressure rise will occur and at the initial upstream side a pressure drop will occur. Repetitive slamming may occur if vapour pressure is reached at the initial upstream side of the valve.

7 ACKNOWLEDGEMENTS

The measurements concerning the moving ball type non-return valve form part of the measurements carried out for such a valve provided with and without a special anti-slamming device. These tests were carried out on request of Hillen de Lelie B.V. The author thanks Hillen de Lelie B.V. for permission to publish some results of these measurements in this paper.

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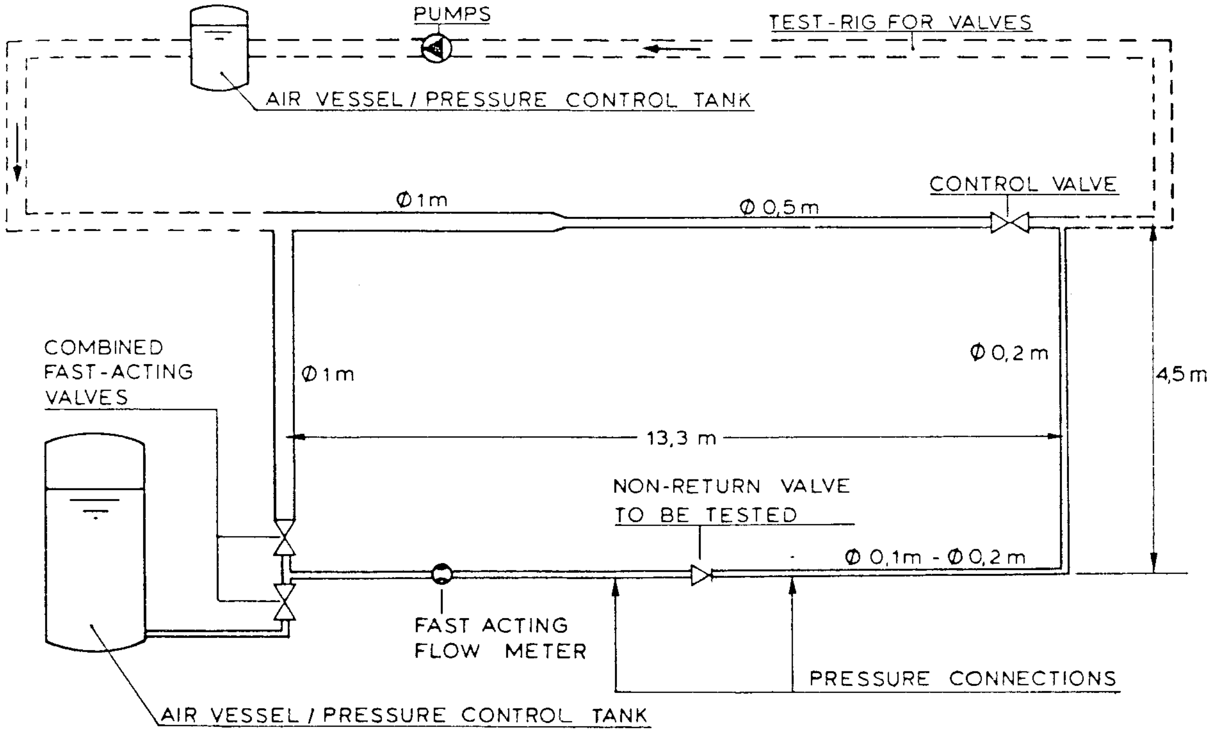


Figure 1 Schematic diagram of test-rig for non-return valves

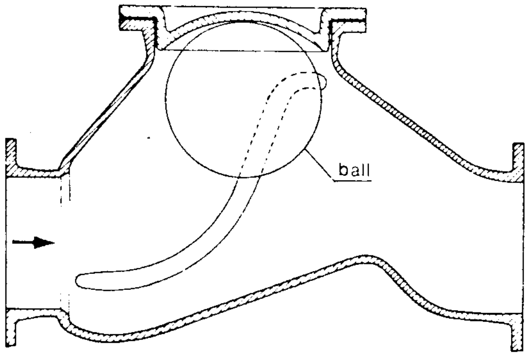


Figure 2 A moving ball type non-return valve

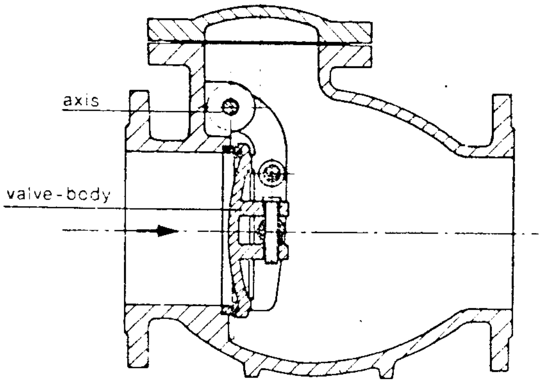


Figure 3 A swing type non-return valve

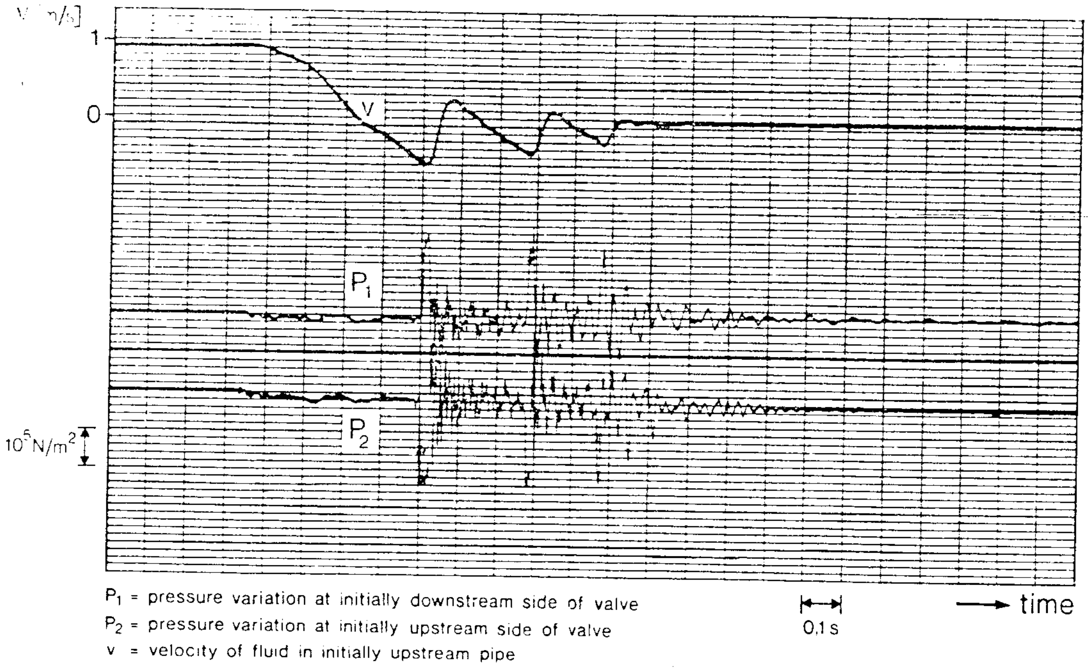


Figure 4 Recording of experiment No. 1
(moving ball type non- return valve)

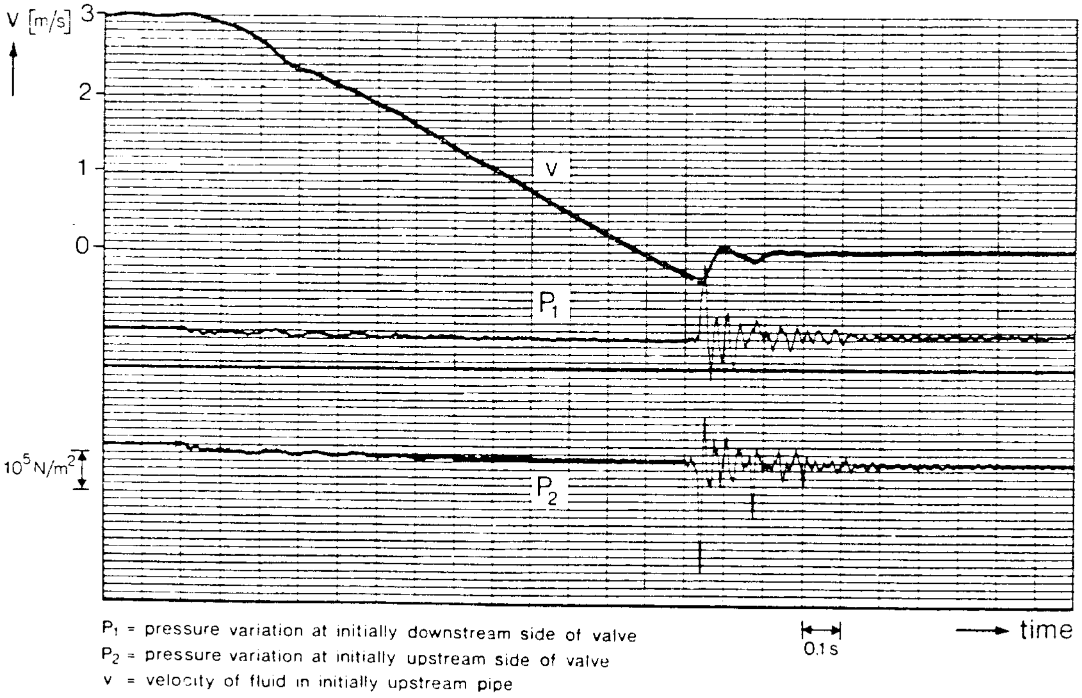
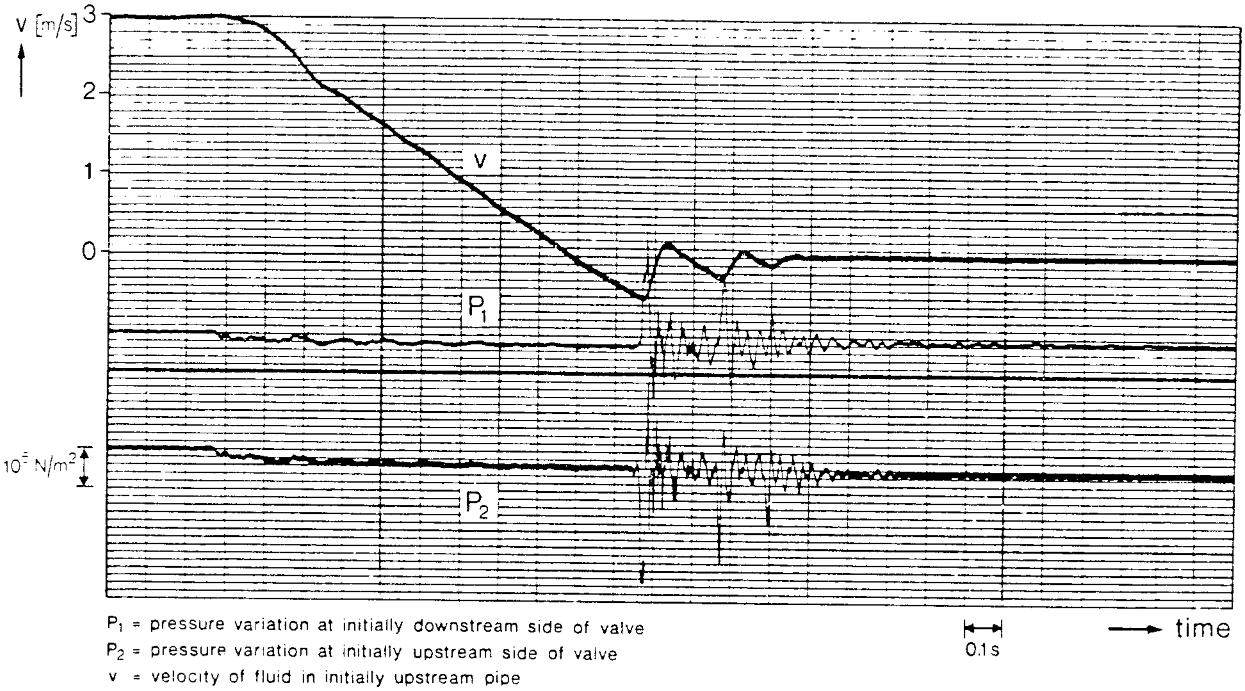
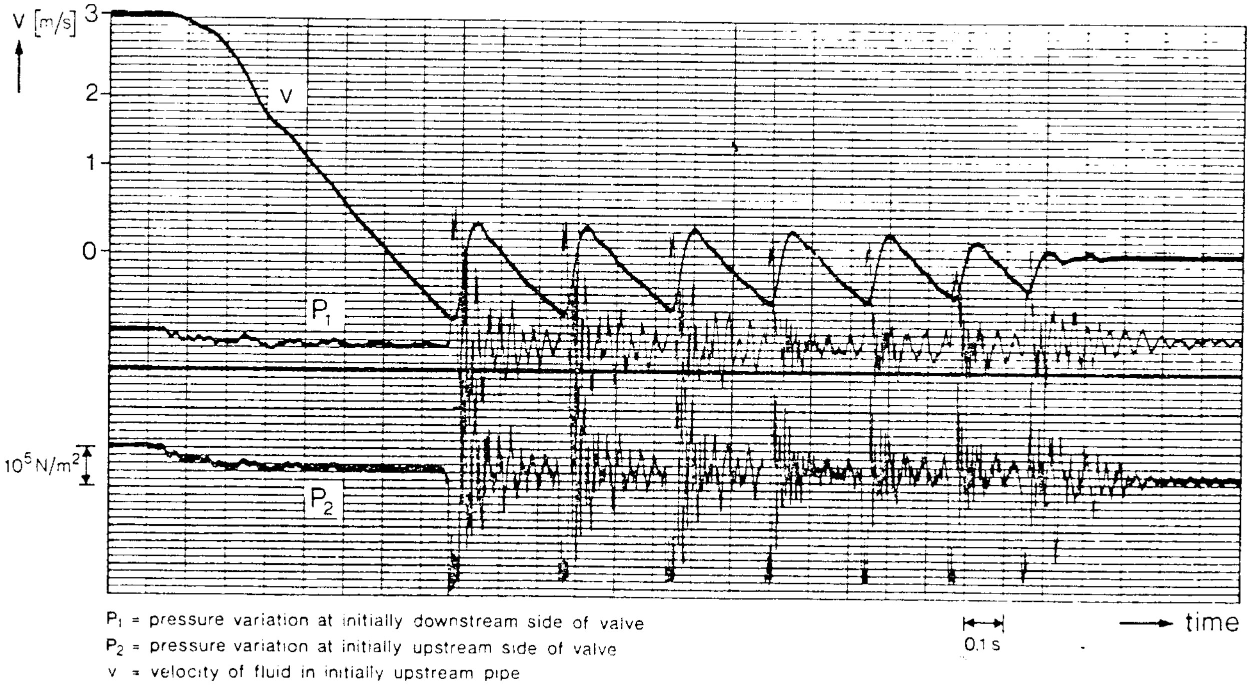


Figure 5 Recording of experiment No. 2
(moving ball type non-return valve)



P_1 = pressure variation at initially downstream side of valve
P_2 = pressure variation at initially upstream side of valve
v = velocity of fluid in initially upstream pipe

Figure 6 Recording of experiment No. 3
(moving ball type non-return valve)



P_1 = pressure variation at initially downstream side of valve
P_2 = pressure variation at initially upstream side of valve
v = velocity of fluid in initially upstream pipe

Figure 7 Recording of experiment No. 4
(moving ball type non-return valve)

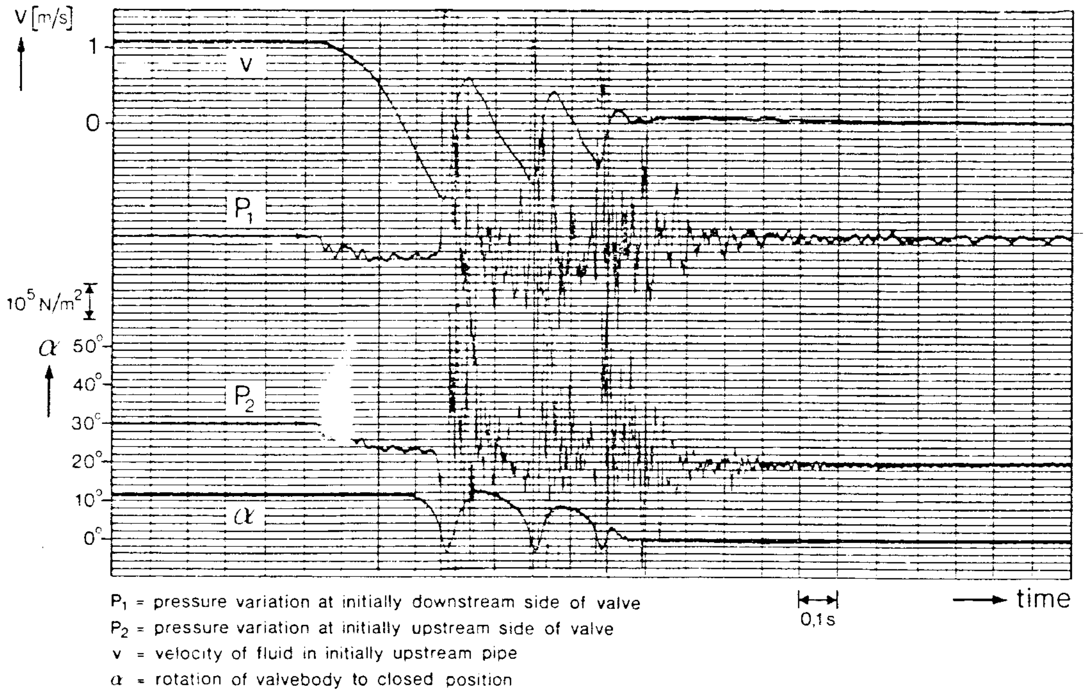


Figure 8 Recording of experiment No. 5
(swing type non-return valve)

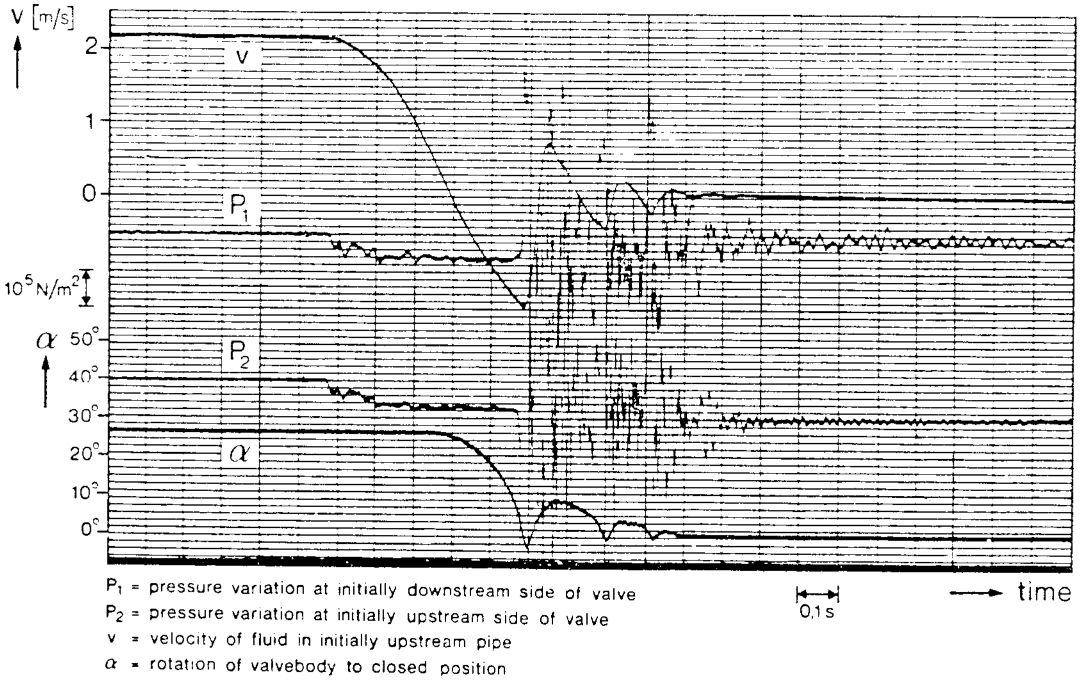
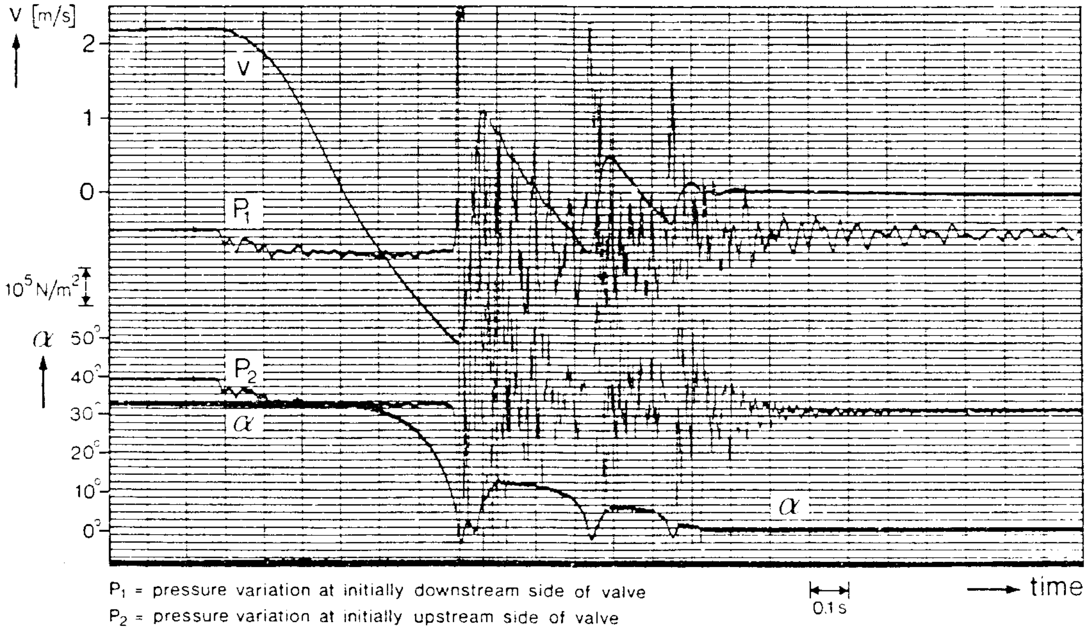
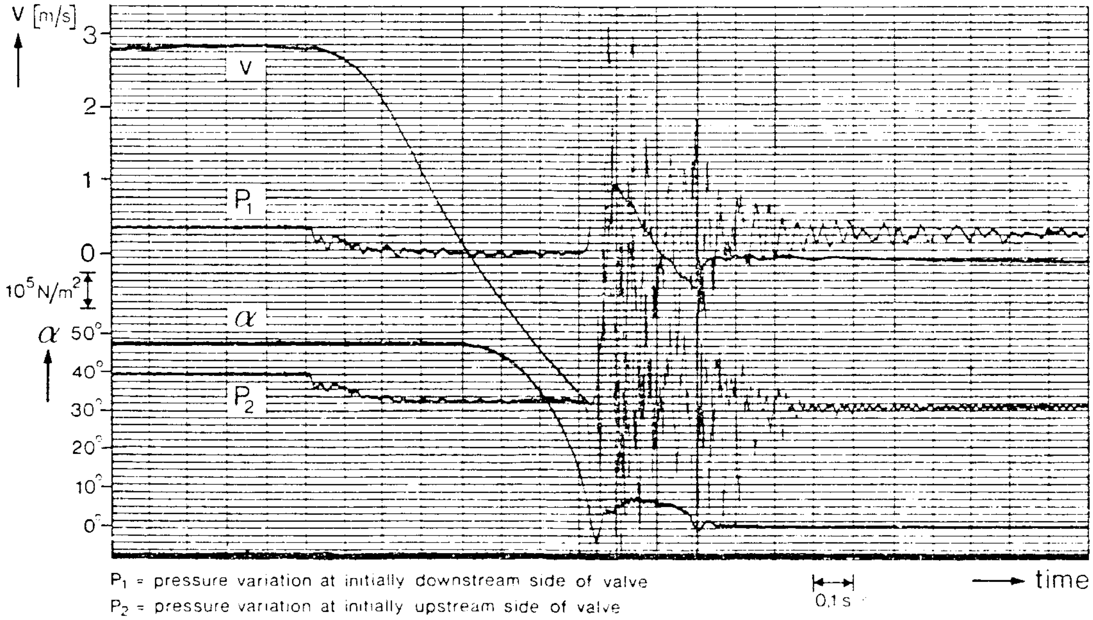


Figure 9 Recording of experiment No. 6
(swing type non-return valve)



P₁ = pressure variation at initially downstream side of valve
P₂ = pressure variation at initially upstream side of valve
v = velocity of fluid in initially upstream pipe
α = rotation of valvebody to closed position

Figure 10 Recording of experiment No. 7
(swing type non-return valve)



P₁ = pressure variation at initially downstream side of valve
P₂ = pressure variation at initially upstream side of valve
v = velocity of fluid in initially upstream pipe
α = rotation of valvebody to closed position

Figure 11 Recording of experiment No. 8
(swing type non-return valve)

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DYNAMIC RESPONSE OF CHECK VALVES

A.R.D. Thorley

The City University
London, England

A badly selected check-valve can be a dangerous addition to a pipeline. Under very rapid flow reversal conditions the pressure rise following ultimate closure of the valve can be disastrous. This paper culls information from various published and unpublished sources and attempts to deduce the important features of good valve design. Criteria for aiding the selection of low loss non-slam valves are suggested.

1. INTRODUCTION

An essential component of most pipeline systems is the non-return valve - also known as the check valve or reflux valve. Its usual functions are to limit backward flow through, and reverse rotation of, pumps and their driving motors and to prevent the emptying of reservoirs.

In principle their construction is simple, comprising a moving element, or elements, which is/are controlled primarily by the fluid. Historically, the most common device is the simple swing check valve.

In valves such as these the mean velocity of the fluid is backward at the instant when the valve shuts. The large pressure rises associated with the reduction of this reverse velocity accelerate the closure, to give yet higher pressures which ultimately slam the moving element onto its seat.

This phenomenon of 'check valve slam' has assumed increasing importance in recent years leading to various test programmes, modifications to existing valves and the development of new styles of valve (1). At the present time there exists some degree of confusion over the most appropriate remedies for the problem of valve slam and the criteria that should be applied when selecting the most suitable check valve for a particular application.

Being an unsteady turbulent flow problem developments have followed the two traditional paths of fluid mechanics; on the one hand there have been the quasi-theoretical approaches requiring ultimately some experimental coefficients, whilst on the other there has been the almost entirely intuitive and empirical trial-and-error evolution of the 'non-slam' valves.

2. THEORETICAL STUDIES

When a check valve slams shut in response to a reverse flow the sudden pressure rise Δp generated may be predicted from Joukowsky's well known equation

$$\Delta p = \rho c \Delta V \qquad (1)$$

where c is the speed of propagation of pressure waves through the fluid of density ρ and ΔV is the magnitude of the velocity change. As the closure is normally 'rapid' in terms of the pipeline period ΔV has the same magnitude as the maximum reverse velocity attained by the fluid during valve closure.

To evaluate equation (1) or to include the check valve as a boundary condition (other than a simple open/shut device) in a numerical solution by computer it is necessary to consider the dynamic response of the valve and obtain a suitable relationship for dv/dt which can be integrated over successive time intervals.

In two early complementary studies of 50 mm (2 inch) swing check valves Worster developed a linearized theory for providing quick estimates of valve door response. Linearising the equation of motion into a form that is easily integrated leads to analytical expressions for the variation of valve angle with time and the time lag between the reversal of water flow and the closing of the valve. Comparisons with experimental data suggested that the inertia of the valve door had very little effect, but that pin friction and the displacement volume swept out by the valve were much more important. Further tests on larger valves were recommended.

More recent studies (4, 5) also for swing check valves considered component torques separately in the equation of motion, viz.

$$T_G + T_e - T_f - T_B - T_P = (I_D + I_W)\,\ddot{\theta} \qquad (2)$$

These components are due to gravity (T_G), externally applied torques (T_e), friction (T_f), buoyancy (T_B) and hydrodynamic pressure forces (T_p). The inertia comprises a component (I_D) due to the valve door itself plus an additional component (I_W) representing an added mass effect. θ is the angular acceleration of the valve door.

The gravitational, buoyancy and externally applied torques are easily determined from valve dimensions and material properties. It was speculated (5) that the friction torque would be of the form

$$T_f = K_1 + K_2\,\ddot{\theta}^n \qquad (3)$$

in which K_1 and K_2 are constants and n an arbitrary but constant power. However experimental results suggested that an additional term ($-K_3\,\ddot{\theta}^m$) be included representing the rate of change of acceleration.

The hydrodynamic torque T_p will depend on the flow pattern around the valve, its position and the flowrate Q i.e.

$$T_p = f(Q, \theta, \dot{\theta}) \qquad (4)$$

The precise form of the relationship cannot be found analytically - a probable form might be supposed and empirical coefficients determined for particular examples.

Of the two components of the inertia, I_D, the inertia of the disc may be readily calculated knowing valve dimensions. The added mass effect is speculative, but from experiments in air and in water to determine the periodic time of small oscillations of a 150 mm valve door it appears (5) that the added mass effect could be estimated as being equivalent to a sphere of water having a diameter the same as the valve door and with their geometric centres coincident.

Plug type check valves for use in power stations have been studied theoretically by Kubie (6) who considered an equation of motion, similar to equation (2), for the linear movement of the valve plug. The valve was assumed to behave as a variable area orifice; frictional and added mass effects were assumed negligible, but an allowance for entrapped air was included. Numerical studies of a typical power application yielded plausible results but experimental studies are required for comparison.

With more general applications in mind Provoost (4, 7) and Collier and Hoerner (8) have focussed their attention on (dv/dt), the rate of change of velocity of the fluid adjacent to the valve. Recognising that check valves interact with the system of which they are a part they speculate on a suitable characteristic that could be used to aid check valve selection. Assuming that, at least for simple pipe systems, one can apply rigid column theory as a first approximation the equation of motion for the liquid column, neglecting friction, has the form:

$$\frac{\Delta H}{L} = \frac{1}{g} \cdot \frac{dV}{dt} \qquad (5)$$

Inserting the best available value of the head difference ΔH (static head for example) across the pipeline of length L yields a mean value for the deceleration of the liquid column. Using this result in conjunction with charts based on experimental studies a valve having a suitable response characteristic can be selected.

These and other experimental studies are discussed in the next section. However it is clear in all the theoretical studies encountered that very little progress has been made without resort at some stage to empirical or assumed information.

3. EXPERIMENTAL STUDIES

Since check valves interact with the system of which they are a part, care must be exercised in interpretating experimental data for transient response. In the first instance it is optimistic to expect more than comparative results for various valves tested in a particular system.

Collier and Hoerner (8) report an experimental study of the split disk type of butterfly valve and a swing check valve, both 150 mm (6 inch) nominal diameter. The split disk valves were spring assisted and two different spring strengths were used. Results are plotted as water hammer pressure versus liquid column deceleration, the latter being rendered dimensionless by dividing through by the acceleration due to gravity. Tests were conducted in both horizontal and vertical pipelines.

The results demonstrated that the pressure rises associated with the closure of the split disk valve were less than those with the conventional swing check valve in the line. Furthermore, the strength of the return springs in the split disk valve was very significant - stronger springs gave rise to a much lower slam pressure. Mounting both valves vertically and without springs in the split disk valve gave results having a similar pattern.

Following a similar philosophy, Provoost (4, 7) independently reports tests on moving ball and swing check valves, expressing the data as maximum reversed flow velocity V_R versus liquid column deceleration - essentially the same as Collier and Hoerner (8) since $\Delta p \; \alpha \; V_R$. Typical flow characteristic - time records are shown (from (1) and (4))in Figures 1 and 2.

Collecting from several sources (1) data that to some degree lend themselves to comparison and displaying in a Provoost-style dynamic characteristic forms the basis of Figure 3. Clearly there are significant differences between the four types of valve represented - in terms of size, style and spring strength. The latter is important but it is not the only design parameter that can be varied as will be discussed later. For the moment this diagram, Fig.3 provides one of several comparisons that can be made. It should be noted that the valves are typical off-the-shelf specimens, no particular care being taken to match their performance characteristics to the test system.

Earlier experimental studies (1, 9) of the swing check valve yielded the so-called 'recoil' or 'non-slam' valve. The principal design changes introduced to achieve this were

(a) to angle the valve seat so that it was inclined in the normal direction of the flow,

(b) a restriction on the valve opening, to as little as 25° maximum in some cases, and

(c) an increase in the closing moment.

The argument for the first two changes is that by reducing the distance through which the valve door must move closure will occur more quickly, especially if assisted, as under (c). Some manufacturers have increased the mass of the door and/or added external levers and weights, whereas others have introduced springs. Spring assistance has the advantage that it does not affect the inertia of the moving door, and in some cases incorrect mounting of additional weights has been counter-productive. Nevertheless, the intelligent use of these modifications can be beneficial, especially in pipelines which are not subjected to very rapid decelerations of the liquid column.

A set of pressure-time histories (Figs.4-6) for a series of tests comparing swing check, split disk and a plate check valves having a nominal diameter of 250 mm are reported (10). The tests were conducted on a simple test rig designed to simulate a pump trip. No data on the velocity - time history of the fluid is available and inclusion on Fig.3 is not possible. The three valve types yielded successively better performances, the swing check being the worst and the plate valve being the best. The closure of both the split disk and plate valves was spring assisted. This requires further discussion as with the question of valve movement.

Perhaps the most arduous conditions that a check valve must withstand are when one of two or more pumps running in parallel is tripped out. As a result of major operating difficulties some plant operators have found it desirable to undertake comparative tests.

Due to serious pressure surge problems associated with conventional swing check valves fitted in a cooling water system (11) comprising four pumps in parallel (see Fig.7) one swing check valve was replaced with a nozzle valve.

The pumps are rated at 50 m head when delivering 833 ℓ/s at a speed of 980 rev/min and having a power requirement of 470 kW. The swing check valves were of 600 mm nominal diameter but the replacement nozzle valves were of 500 mm diameter.

Figure 8 is one of several comparisons (11) from a series of pump trips. In this particular example three pumps were running initially and the one for which the p-t traces apply was tripped. The comparisons have to be regarded to some extent as approximate because there was a six month gap between the swing check and nozzle valve tests. Additionally, the pump rotor was turned down from 650 mm to 620 mm diameter. However, the initial steady discharge pressures are of the same order so by implication the flows would be too.

Despite those qualifications on the data the nozzle valves show a considerably better performance under transient conditions and all the swing check valves were replaced.

Another independent study (12) concerned with a power plant condensate extraction system provided comparisons between a counter-weighted swing check valve and a nozzle valve during various unsteady operating conditions. Figures 9 and 10 are one of several sets of comparisons - in this case dealing with a manual pump changeover. The sequence in both cases is:

Pump 'B' running - Start Pump A - Trip out Pump B.

The pressure-time recordings (retraced for clarity) are not quite aligned vertically, but the upper trace in both sets covers the start-up of Pump A and the lower one records the trip-out of Pump B. The performance of the nozzle valves was found to be considerably better than the counter-weighted swing check valves. When the latter had been replaced it was recommended that the previously imposed operating restrictions be removed.

4. INTERPRETATION OF RESULTS

Despite the results presented above and in Reference 1 having various origins there are sufficient comparisons of different valves in similar pipe systems to enable some deductions to be made.

The most severe operating conditions are those involving the tripping out of one pump from two or more running in parallel, especially in high head systems. Although for some applications such as single pump low head rising mains the conventional swing check valve may be adequate it is not so suitable for arduous duty, even with modifications. Response - time histories (4) illustrate how the flow can have reversed before the valve door even begins to move.

More rapid responses to changing flow conditions are obtained if the moving parts of the valve are of low inertia and are assisted by a spring or springs. Closure times will be shorter and reverse velocities minimised if the distance/angle through which the moving elements have to travel are small.

Reducing the valve opening may tend to increase the frictional head loss. Figure 11 is a comparative plot of the non-dimensionless loss coefficient K (= $2g\Delta H/v^2$) versus the flow Reynolds number for various valve types. The Reynolds number Re is based on the mean velocity V of a fluid of kinematic viscosity ν flowing in a pipe having a diameter D equivalent to the nominal diameter of the valve. These particular curves are for valves having a nominal diameter of 250 mm and are based (1) on reliable experimental data. Adding springs or levers and weights will tend to cause the left hand ends of the curves to rise - quite steeply in most cases - as shown for a typical weighted check valve which in this case is not fully open until a Reynolds Number of 8×10^5.

The spring assisted valves can be most easily 'tuned' to the system of which they are a part. The spring strengths should be so specified that the valves are fully open by, say, 70% of their normal flowrate.

Plate and nozzle valves both have short travel distances for the moving elements but the streamlined shape of the latter gives a superior pressure recovery Whilst the valve loss may be small compared to the overall losses in the system one should, in an energy conscious world, strive to make savings however small.

With regard to the mathematical modelling the great range of valve types and the variability of component specifications means that the development of an accurate and general model is unlikely. However, a quasi-theoretical approach, possible similar to the Provoost-Collier-Hoerner scheme will provide a useful basis for the comparison of generic types.

5. CONCLUSIONS

For arduous conditions of service such as those associated with high head multi-pump duty it is suggested that in the interests of avoiding check valve slam the following criteria should be applied when selecting a suitable non-return valve:

(1) low inertia of the moving parts

(2) short travel distance/angle

(3) air- or coil-spring assistance of closure, commencing at, say, 70% of the rated flow.

Further controlled tests are required to compare different valve types having previously tuned each one to give its optimum performance in the selected test facility.

6. ACKNOWLEDGEMENTS

Experimental data were provided by several organisations and their permission to publish is acknowledged as follows. Figures 2 and 3 contain data provided by Mannesmann Demag; Figure 3 also contains data provided by Gestra (UK) Ltd. The original experimental work was undertaken by the Delft Hydraulics Laboratory in The Netherlands.

The original data for Figures 4-6 were provided by Alsthom-Atlantique of Grenoble whose assistance is much appreciated and APC Ltd. kindly gave permission for the inclusion of Figures 9 and 10.

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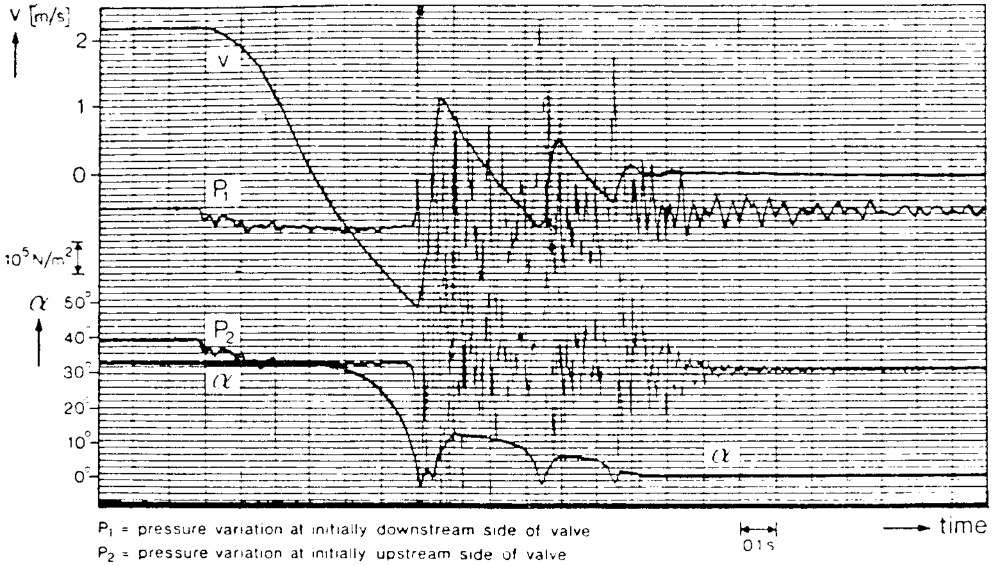

P_1 = pressure variation at initially downstream side of valve
P_2 = pressure variation at initially upstream side of valve
v = velocity of fluid in initially upstream pipe
α = rotation of valvebody to closed position

Fig. 1 Velocity, pressure and position-time curves for a swing check valve - mean liquid deceleration = 6.7 m/s^2 (from Ref.4).

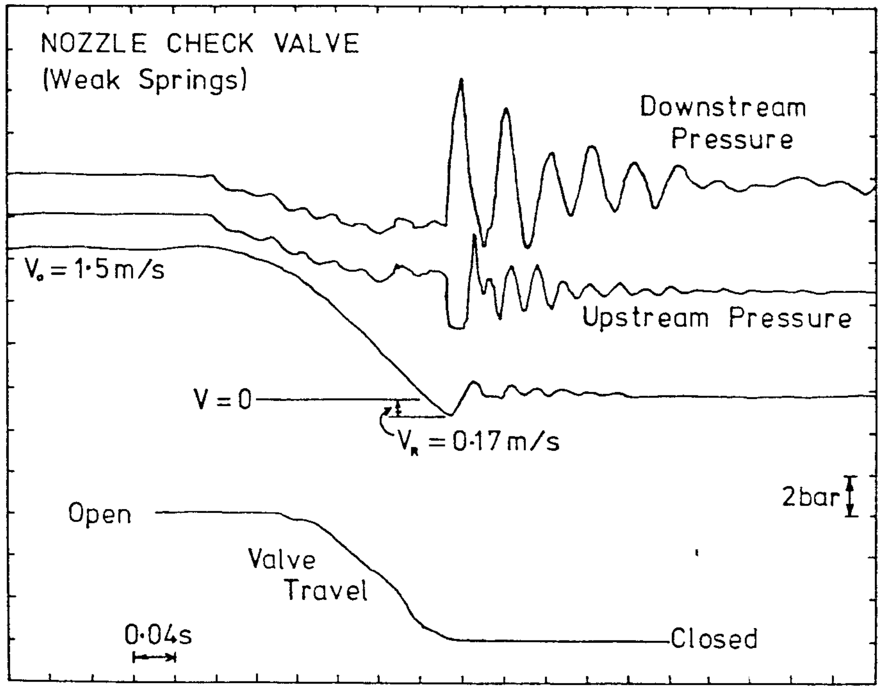

Fig. 2 Velocity, pressure and position-time curves for a nozzle check valve (from Ref.1).

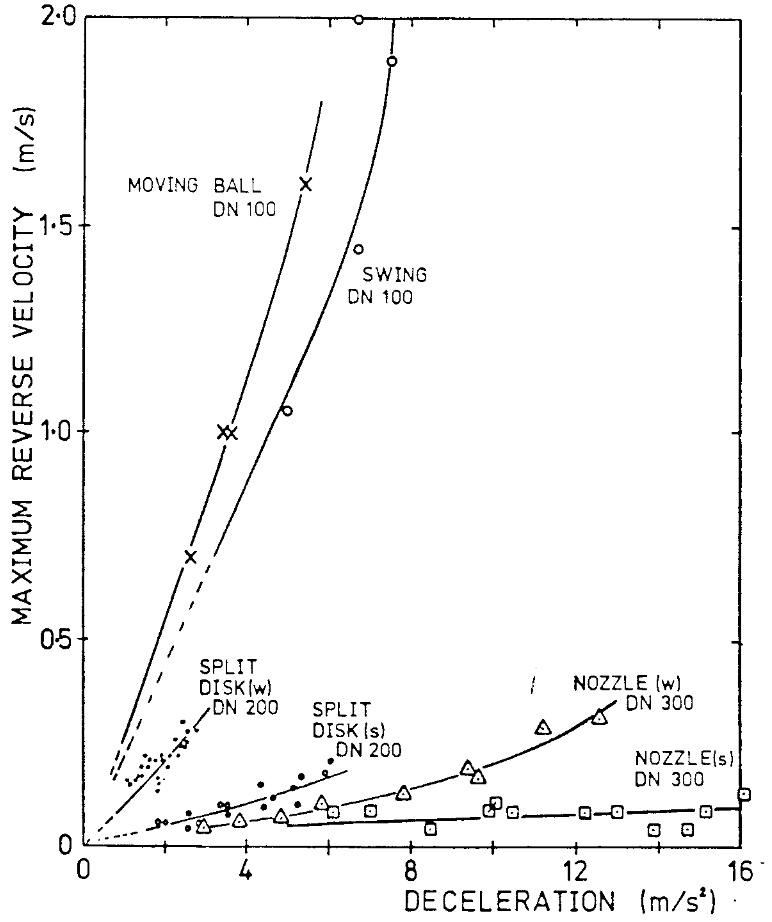

Fig. 3 Maximum reverse velocity as a function of water column deceleration for various valve types. (s) and (w) refer to data for valves fitted with strong and weak springs respectively.

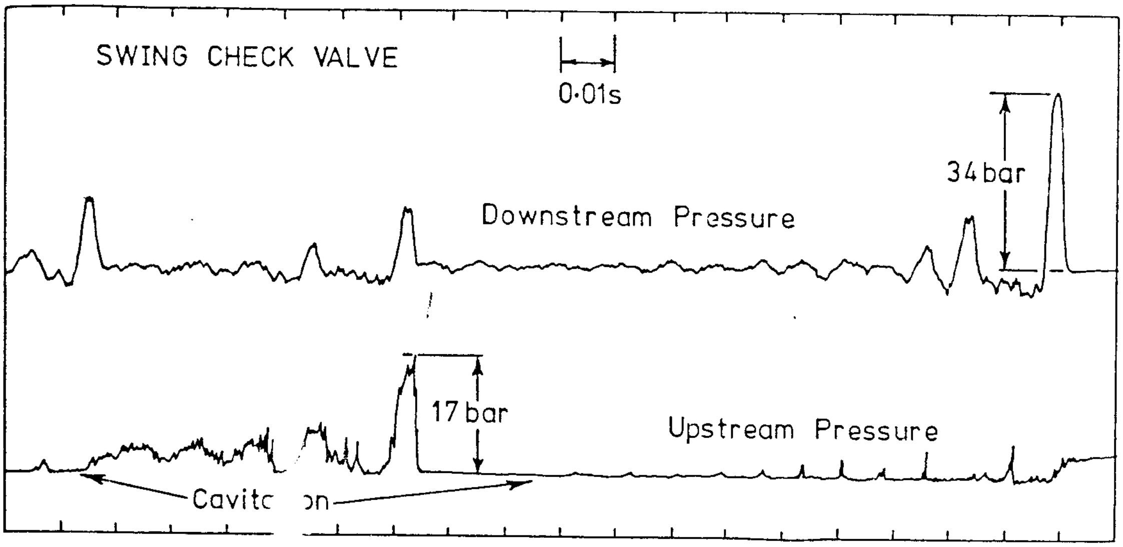

Fig. 4 Pressure-time histories upstream and downstream of a swing check valve following a simulated pump trip.

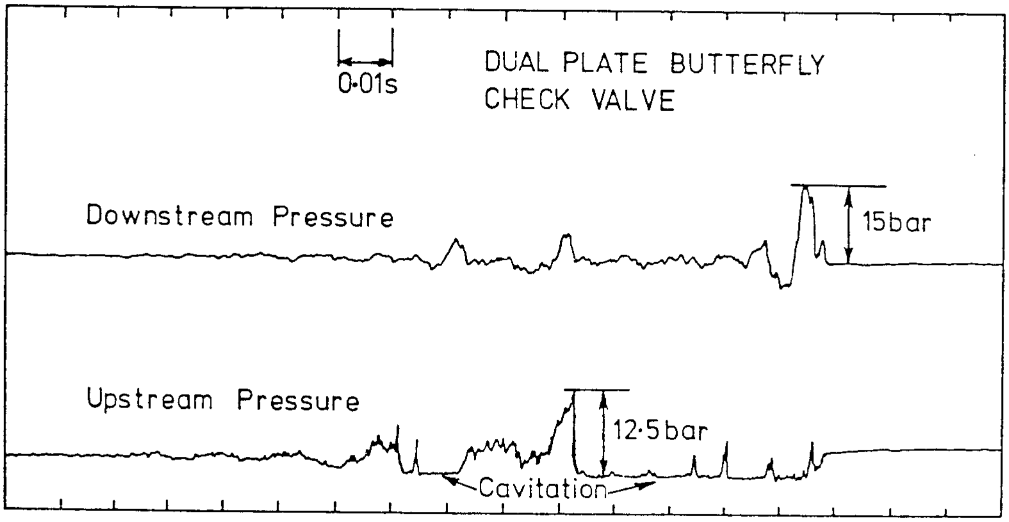

Fig. 5 As Fig.4 but for a dual plate butterfly check valve.

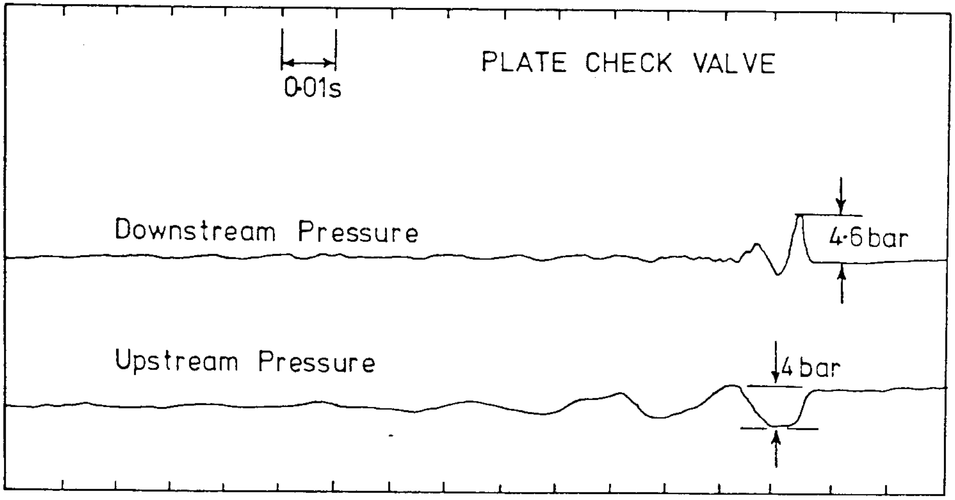

Fig. 6 As Fig.4 but for a plate check valve.

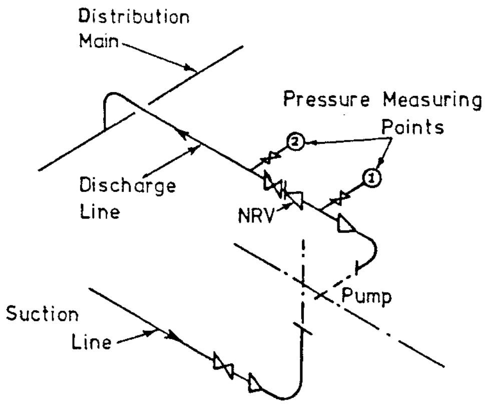

Fig. 7 Layout of one of four similar pumps showing the location of the valves and pressure measuring points.

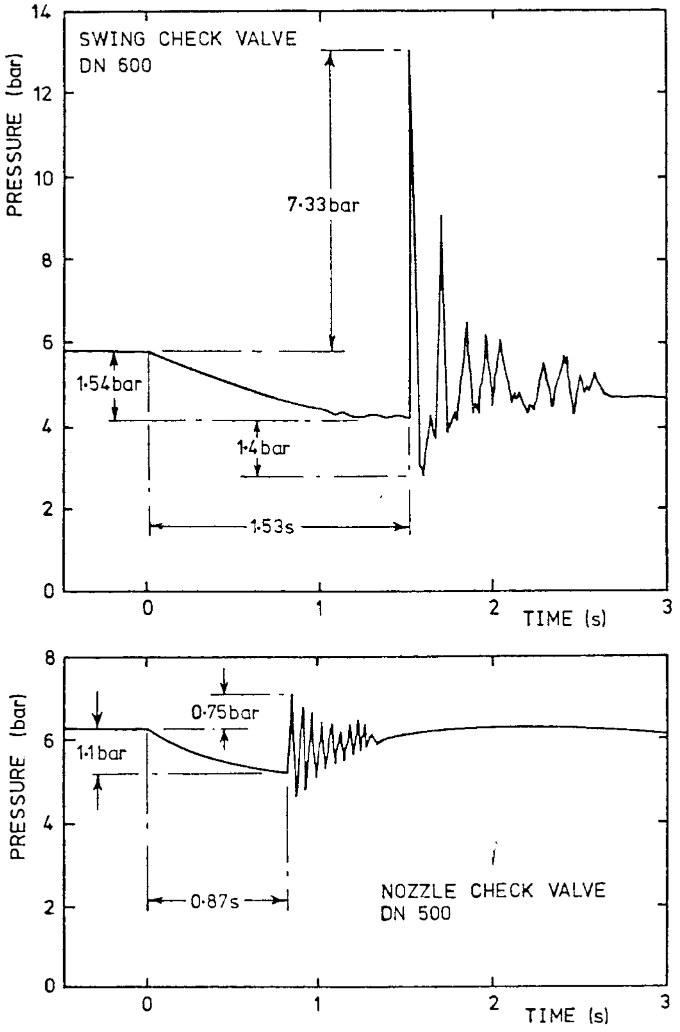

Fig. 8 Comparison of pressure-time histories immediately downstream of the check valve (point 2). Three pumps were in service initially, this one tripped out at zero time.

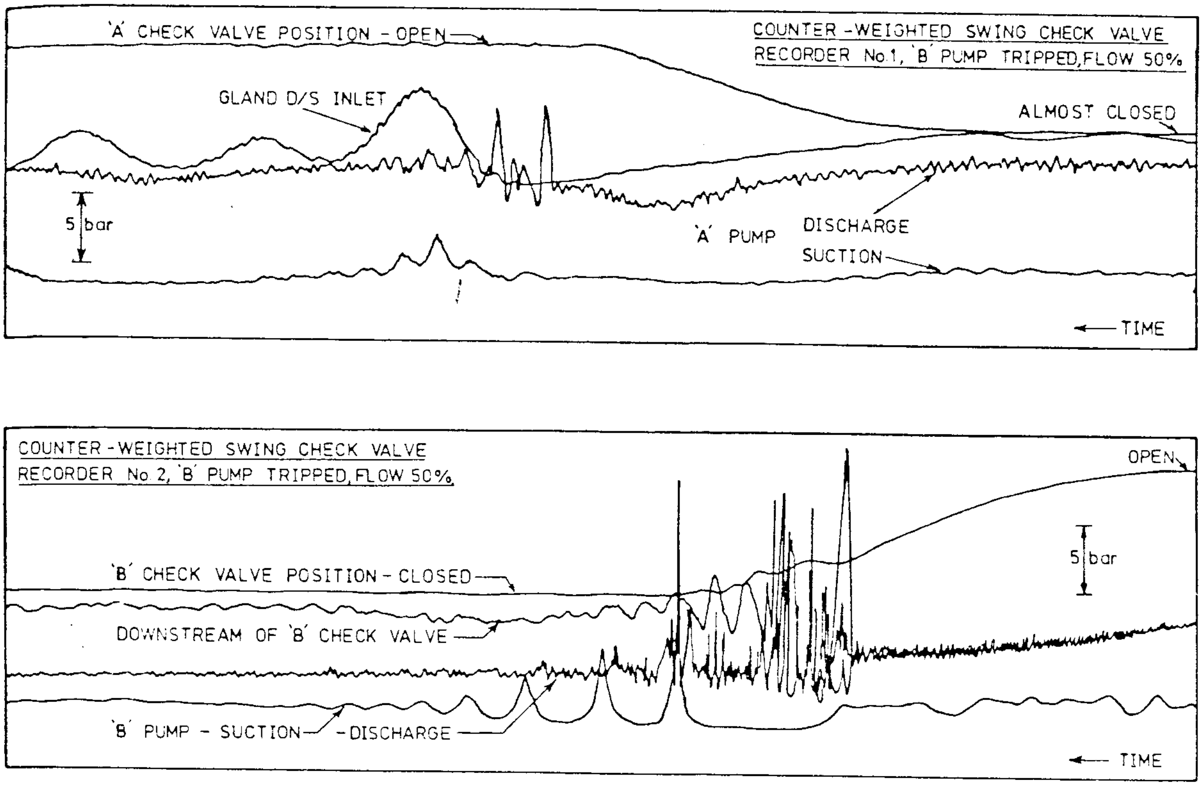

Fig. 9 Pressure-time records for a manual changeover from Pump A (upper trace) to Pump B (lower trace) - both pumps fitted with counter-weighted swing check valves.

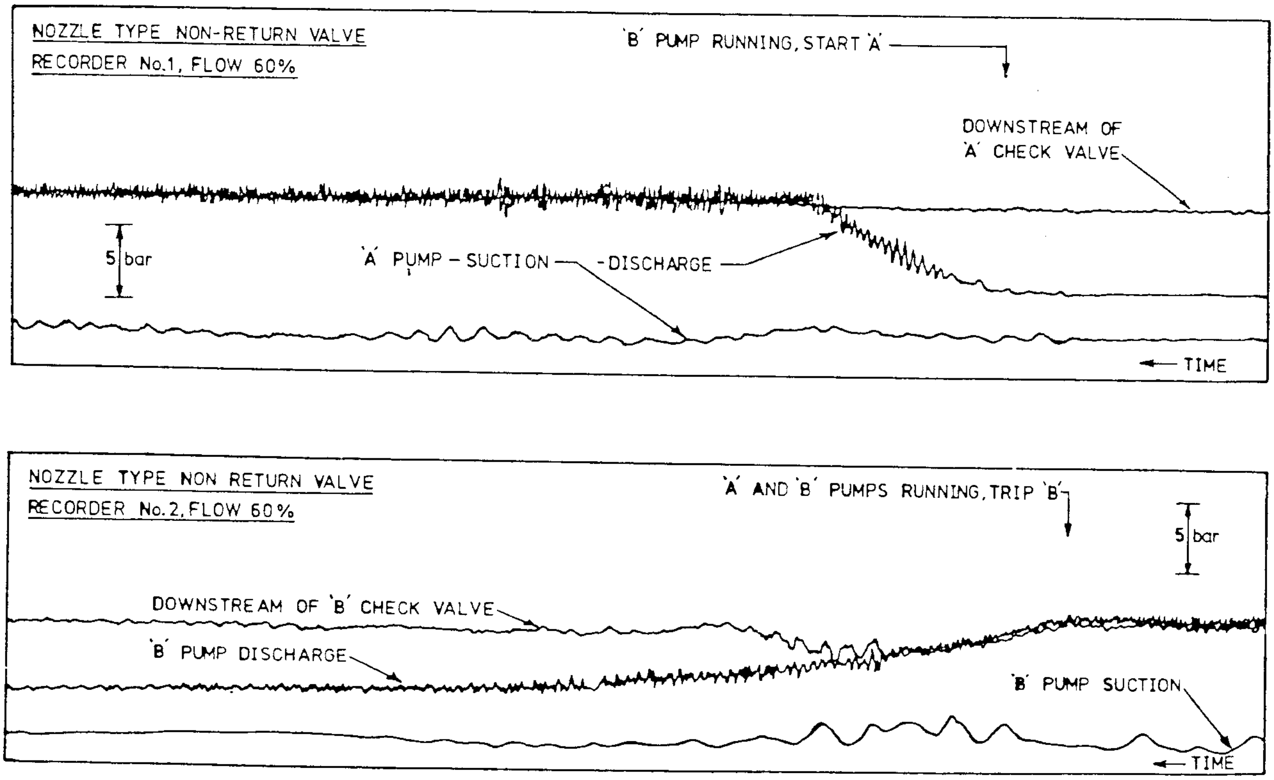

Fig. 10 Pressure-time records for a manual changeover from Pump A (upper trace) to Pump B (lower trace) - both pumps fitted with nozzle style check valves.

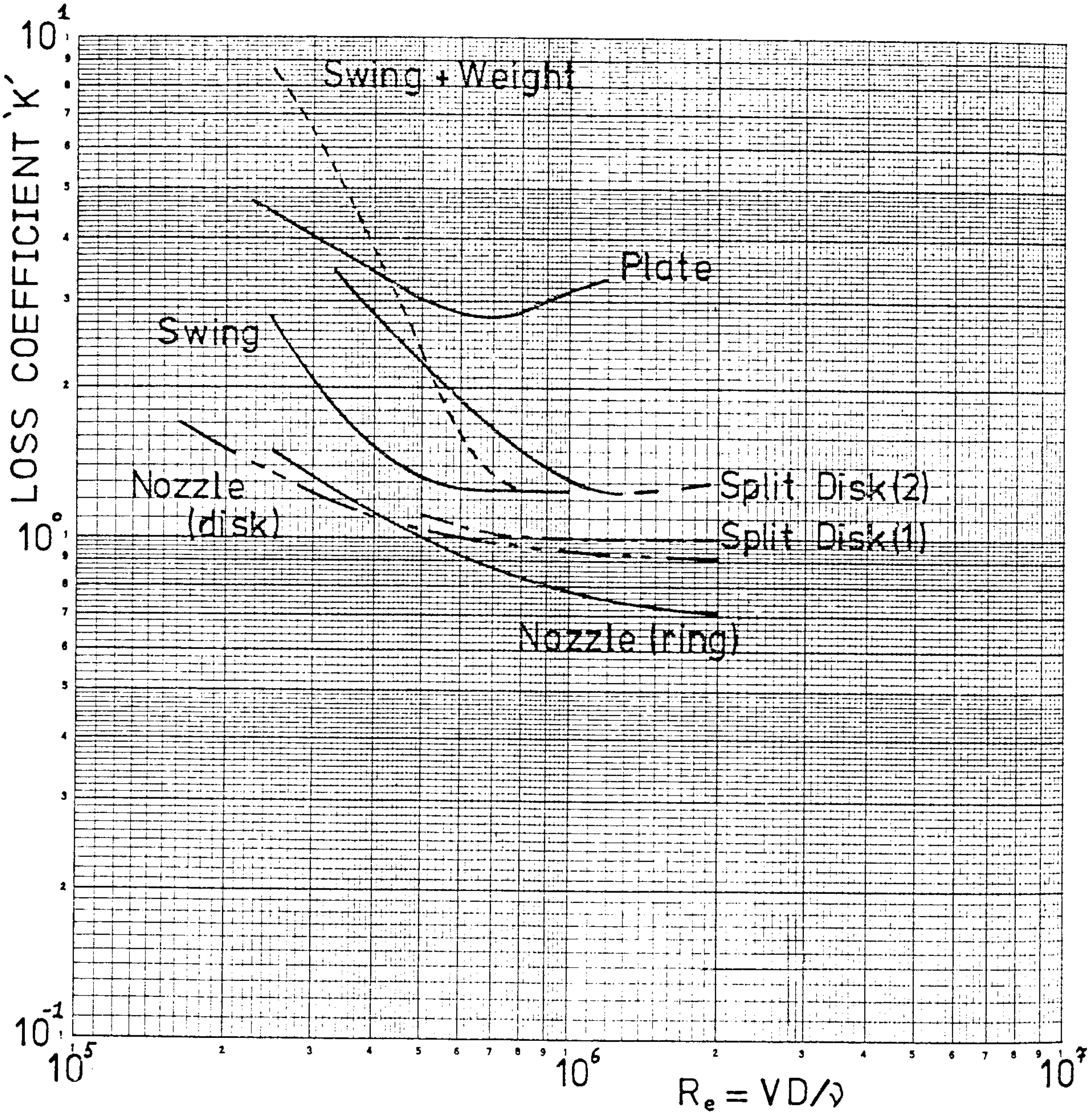

Fig. 11 Comparison of loss coefficients for 250 mm diameter check valves of different designs.

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PAPER F4

A CRITICAL ANALYSIS TO DETERMINE DYNAMIC CHARACTERISTICS OF NON-RETURN VALVES

G.A. Provoost

Delft Hydraulics Laboratory
P.O. Box 177
2600 MH DELFT
The Netherlands

Summary

Non-return valves serve to prevent the reversal of flow. In a quickly decelerating flow they may close during reversed flow. For undampened valves a sudden decrease of the reversed flow to zero will occur at closure, possibly resulting in unallowable pressures and slamming of the valve.

The dynamic property of undampened non-return valves can be presented in the form of a dynamic characteristic. This characteristic contains the relationship between the reversed liquid velocity at closure and the deceleration of the liquid passing the valve.

The measured dynamic characteristic is shown for several types of valves. Also an analytical approach to determine the dynamic characteristic is dealt with. Such an approach gives insight into the relevant parameters and may be helpful to determine the relevant range of decelerations to be measured.

NOMENCLATURE

A = area of moving part of valve (m^2)

a = wave speed (m/s)

D = diameter of valve body (m)

F_e = force due to external loads (N)

F_f = force due to friction (n)

F_h = force due to hydrodynamic pressures (N)

F_I = force due to inertia (N)

F_w = force due to weight of moving part (N)

k_F = hydrodynamic force coefficient ($-$)

L = length of connecting pipe (m)

Δp = pressure difference (N/m^2)

Q = flow rate passing the valve (m^3/s)

V_D = volume displaced by moving part (m^3)

v = liquid velocity (m/s)

v_R = reversed liquid velocity (m/s)

x = displacement of moving part (m)

t = time (s)

1. INTRODUCTION

Non-return valves serve to prevent reversal of flow. If, however, reversal of flow occurs quickly, most valves will close after the flow has already been reversed. From former tests (Ref. 1) it was deduced that the motion of decelerating flow is hardly affected by an undampened non-return valve until the valve closes, see Fig. 2.

Dampened non-return valves will generally have a considerable effect on reversed flow, see also Fig. 1. A well adjusted damper will improve the phenomena after valve closure. A badly adjusted damper however may lead to worse dynamic phenomena. These types of non-return valves will not be considered here.

This paper only deals with undampened non-return valves. These valves have a free moving part such as a disc the motion of which is only influenced by the motion of the liquid, weight or spring. In decelerating flow the moving part will move to the closed position. When the moving part of the valve hits its seat an almost instantaneous decrease of the reversed flow to zero will occur, hence resulting in a sudden pressure rise at the initial downstream side of the valve and a sudden drop at the initial upstream side.

2. THEORY

The following analysis concerns valves with a translating moving part. This consideration, however, is also applicable to valves with rotating parts. The motion of the moving part is described by Newton's Second Law of Motion:

$$F_I + F_w + F_e + F_f + F_h = 0 \qquad (1)$$

The terms in this equation represent respectively the forces exerted on the moving part due to inertia, weight, external forces, friction and hydrodynamic pressures. The inertia term is $m\,\ddot{x}$ where m is the mass of the moving part and surrounding fluid and x is the position of the moving part. The force due to hydrodynamic pressures depends on the velocity profile inside the valve, and position and motion of the moving part:

$$F_h = f(Q, x, \dot{x}, \ddot{x}) \qquad (2)$$

For reasons of simplicity the effect of motion of the moving part of the valve is considered by assuming the flow rate passing the moving part to be equal to the flow rate in the connection pipes from which the flow rate induced by the displacement of the moving part is substracted.

This kind of analysis was also followed by other investigators (Refs. 2, 3, 4 and 5). The force due to hydrodynamic pressures is then assumed to be proportional to the flow rate passing the moving part squared:

$$F_h = \frac{\rho k_F}{D^2} (Q - A\,\dot{x})^2 \qquad (3)$$

In this equation the hydrodynamic coefficient k_F is a dimensionless coefficient, which is assumed to be solely dependent on the position of the moving part. The dependence between k_F and the relative motion of the moving part $(A\dot{x}/Q)$ mentioned in Ref. 6 is neglected. An experimental determination of this coefficient is preferable (Ref. 3) but is not always possible.

The motion of the moving part can be determined for a given time history of the flow passing the valve with the equations (1) and (3) completed with specific relationships for the forces due to weight, external loads and friction. From this result the reversed flow rate and reversed liquid velocity $v_{R_{max}}$ at the instant of closure is obtained. Once the valve is closed a sudden decrease of the reversed flow to zero will occur, resulting in pressure variations at both sides of the valve. The maximum possible value of these pressure variations is given by:

$$\Delta p_{max} = \rho\, a\, v_{R_{max}} \qquad (4)$$

At the initial downstream side a pressure rise of this amplitude may occur and at the initial upstream side a pressure drop of the same value may occur. The time interval in which the pressure variation is built up is equal to the time interval in which the flow decreases to zero and depends on the elasticity of the connecting pipes, the moving part of the valve and its seat. Whether the maximum possible pressure variation is obtained depends on this time interval and the length of the pipes connecting the valve with reflecting boundaries (i.e. reservoir, air vessel). The maximum pressure variation is reached as long as the time interval is shorter than the wave travel time 2L/a where L is the length of the connecting pipe. If the time interval is longer than 2L/a only part of the maximum possible pressure variation will occur. For a negligible length (for instance when the valve is mounted at the outlet in a reservoir) pressure variation will also be negligible.

The pressure drop at the initial upstream side of the valve can also be limited by the vapour pressure of the liquid. If so, a vapour cavity will be formed, followed by a sudden pressure rise when the cavity disappears. The pressure rise may be of the same order as the maximum pressure variation following from valve closure. Due to this pressure rise the valve may reopen and then close again. In some cases this may lead to hydraulic resonance.

3. DEFINITION OF THE DYNAMIC CHARACTERISTIC

As indicated in Ref. 1 the deceleration dv/dt is of decisive importance for the prediction of $v_{R\max}$. Especially when the valve is fully open in the initial steady condition dv/dt is the only variable determining $v_{R\max}$. So for each valve a graph can be provided showing $v_{R\max}$ as a function of dv/dt. Such a graph characterizes the dynamic property of the valve and was therefore introduced as the dynamic characteristic of non-return valves (Ref. 7). This characteristic depends on the type of valve, dimensions, presence of springs, counterweights, etc. and should be provided by the valve manufacturer, e.g. see Fig. 2.

A similar approach was followed by other investigators (Ref. 8) who gave the pressure rise at the original downstream side as a function of dv/dt, see Fig. 3. The pressure rise depends on the reversed fluid velocity at closure, the elasticity of the moving part and its seat, the wave speed in the connecting pipe and the length of the connecting pipe. Thus this method leads to a graph which depends on the properties of the test facility. The method proposed in Ref. 7 is more general since it is independent of the properties of the test facility as long as the deceleration keeps an almost constant value.

4. THE USE OF THE DYNAMIC CHARACTERISTIC

When choosing a certain type of non-return valve for a pipe system under design or an existing pipe system in which a fast decelerating flow may occur the dynamic behaviour of the non-return valve and resulting pressures should be predicted. Real systems are often very complicated but one can mostly characterize the unsteady flow by the deceleration of the fluid. This deceleration can be determined by calculating the unsteady flow in the pipe system as if there was no non-return valve installed. Once the deceleration dv/dt is known, the maximum reversed liquid velocity $v_{R\max}$ can be found by the use of the dynamic characteristic of the valve of interest. With the aid of Eq. (4) the maximum possible pressure variations at both sides of the valve can be calculated. Together with the predicted pressure at the valve at closure, the maximum and minimum possible pressures at both sides of the valve can be found. It is also possible to programme the dynamic behaviour of a non-return valve in a transient computer code in such a way that the valve closes at a reverse liquid velocity which depends on the velocity gradient of the flow passing the valve. The valve should not cause a pressure drop during reverse flow. Simulations carried out with a computer which includes the above mentioned description of a non-return valve will give a good idea of the unsteady phenomena after closure.

5. DETERMINATION OF THE DYNAMIC CHARACTERISTIC

The dynamic characteristic of a certain type of non-return valve can be determined by performing experimental dynamic tests in which the flow through the non-return valve is quickly decelerated. Both the fluid deceleration dv/dt and the maximum reversed liquid velocity $v_{R\max}$ at closure have to be measured. These measurements can be performed in special test rigs described in Refs. 1, 3 and 8. A typical test result is shown in Fig. 4, showing traces of fluid velocity and pressures versus time.

Each test gives one point in the $v_{R_{max}}$-dv/dt-plane. Through these test data a line can be drawn which is referred to as the dynamic characteristic. In Fig. 5 the dynamic characteristic is shown for a 200 mm dual flapper type non-return valve type BB16 from Gestra, provided with super strong springs, for horizontal position. In Fig. 6 the dynamic characteristic is shown for a 200 mm translating disc non-return valve type RK56 from Gestra, provided with weak springs, for horizontal position. In Fig. 7 the dynamic characteristic of several valves of different types with a diameter of 200 mm are shown for horizontal position. With the aid of the theory developed the dynamic characteristic is calculated for the 200 mm translating disc valve RK56 from Gestra. For the purpose the hydrodynamic coefficient k_F was estimated and is shown in Fig. 8. The diameter of the disc was 180 mm ($A = 0{,}0254\ m^2$) and the movement of the disc from open to closed position was 38 mm. The mass of the disc including surrounding water was estimated at 10 kg. A number of calculations was carried out starting from a steady state condition in which the valve is fully open and imposing a linearly decreasing flow. A typical result of such a calculation is shown in Fig. 9. Each calculation results in a reversed liquid velocity at closure ($v_{R_{max}}$) belonging to the deceleration (dv/dt) imposed. With a number of calculations the dynamic characteristic can be found. Such characteristic for horizontal position without spring is shown in Fig. 10 (curve A).

If the inertia of the disc is neglected the dynamic characteristic can be determined analytically, see also Ref. 9. In that case the disc will follow the motion of the fluid so the displacement of the disc will be equal to the volume of the liquid which passed the valve after reversal of flow. Assuming a constant deceleration dv/dt this leads at the instant of closure to:

$$V_D = \frac{1}{2}\,\frac{v^2_{R_{max}}A}{|dv/dt|} \qquad (5)$$

where V_D is the volume displaced by the disc from open to closed position. This is a "horizontal" parabola in the $v_{R_{max}}$-dv/dt plane:

$$|v_{R_{max}}| = \sqrt{\frac{2\,V_D|dv/dt|}{A}} \qquad (6)$$

For the valve of interst $V_D = 0{,}000967\ m^3$ and $A = 0{,}0254\ m^2$ which gives a characteristic as also shown in Fig. 10 (curve B). Both characteristics with and without inertia as shown in Fig. 10 have the same shape, the difference is caused by the inertia effect.

In Fig. 11 the calculated and measured dynamic characteristics for the RK56 valve provided with weak springs are compared. The resemblence for decelerations up to 10 m/s^2 is evident. For higher decelerations the calculations seem to give too high values for the reversed liquid velocity at closure. Possibly the assumption that the hydrodynamic coefficient is solely dependent on the disc position does not hold for high decelerations. Calculated dynamic characteristics including both inertia and several spring loads are shown in Fig. 12. From these characteristics it may be concluded that the shape for high decelerations is the same, although $|v_{R_{max}}|$ is smaller for a stronger spring. But the shape in the low deceleration region changes with changing spring load. This effect is also clearly visible for the dynamic characteristic of the BB16 valve provided with superstrong springs (see Fig. 5).

6. REMARKS

Results in Ref. 8 show pressure rise curves versus deceleration which all have the shape of a "vertical" parabola, see also Fig. 3. If the downstream pipe is long enough, the pressure rise is proportional to the reversed liquid velocity at closure according to equation (4). In that case the shape of the curves is the same as the shape of the dynamic characteristic. The shape of these curves as well as the shape of the characteristic shown in Fig. 2 differs from the theoretical one shown in

Fig. 10. Calculations as reported here show that the shape of the curves may change with increasing external load, especially in the region of low decelerations. For instance the dynamic characteristic of the 200 mm dual flapper type non-return valve as shown in Fig. 5 also has the form of a "vertical" parabola in the low decelerations zone ($|dv/dt| < 8\ m/s^2$). When determining the dynamic characteristic it is therefore important to perform measurements over a wide range of decelerations.

This range differs from type to type and also depends on the external load. For instance the low deceleration zone for the RK56 valve provided with a weak spring varies from 0-3 m/s^2, while provided with a strong spring this zone varies from 0-12 m/s^2 (see Fig. 12).

The difference between the shape of the curves in Ref. 8 and Fig. 2 and the theoretical shape might be explained by the low decelerations zone of these valves being larger than the area covered by the measurements. This needs further investigation.

7. CONCLUSIONS

The dynamic property of a non-return valve can be presented in the form of a dynamic characteristic. With such a characteristic it is possible to predict the dynamic phenomena after valve closure and to compare the dynamic property of different types of non-return valves. In this paper several types of 200 mm diameter are compared.

Further it is shown that the dynamic characteristic must be determined over a wide range of decelerations. An analytical analysis may be helpful to determine the relevant range to be measured.

8. ACKNOWLEDGEMENTS

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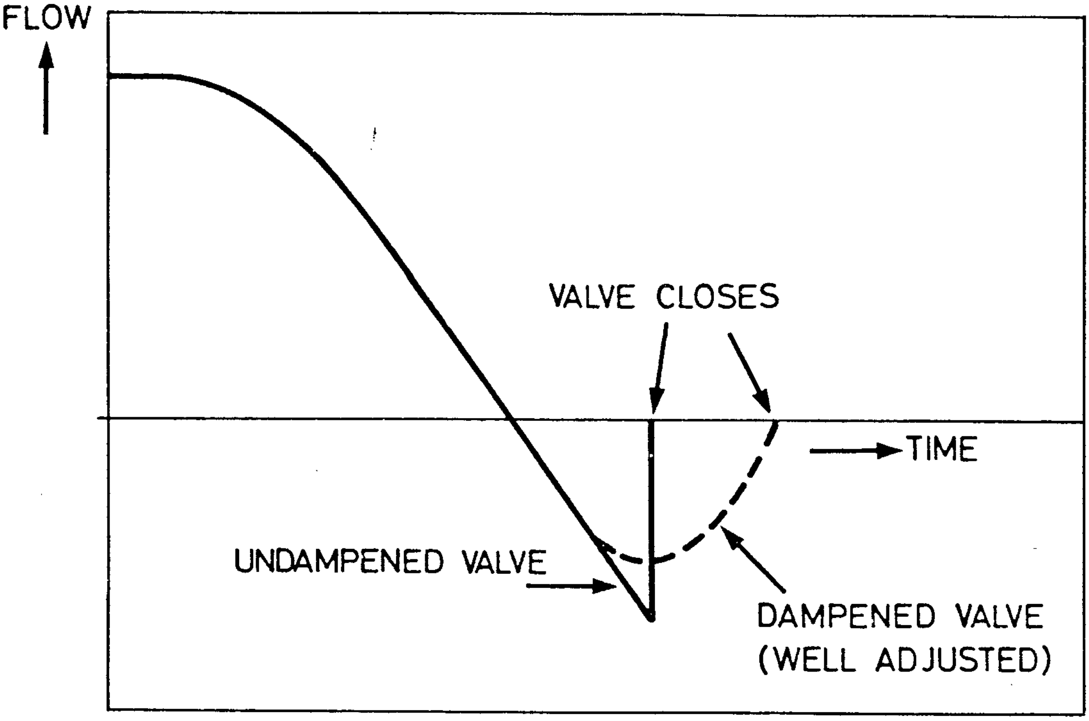

Fig. 1 Flow-time diagram for non-return valves in decelerating flow

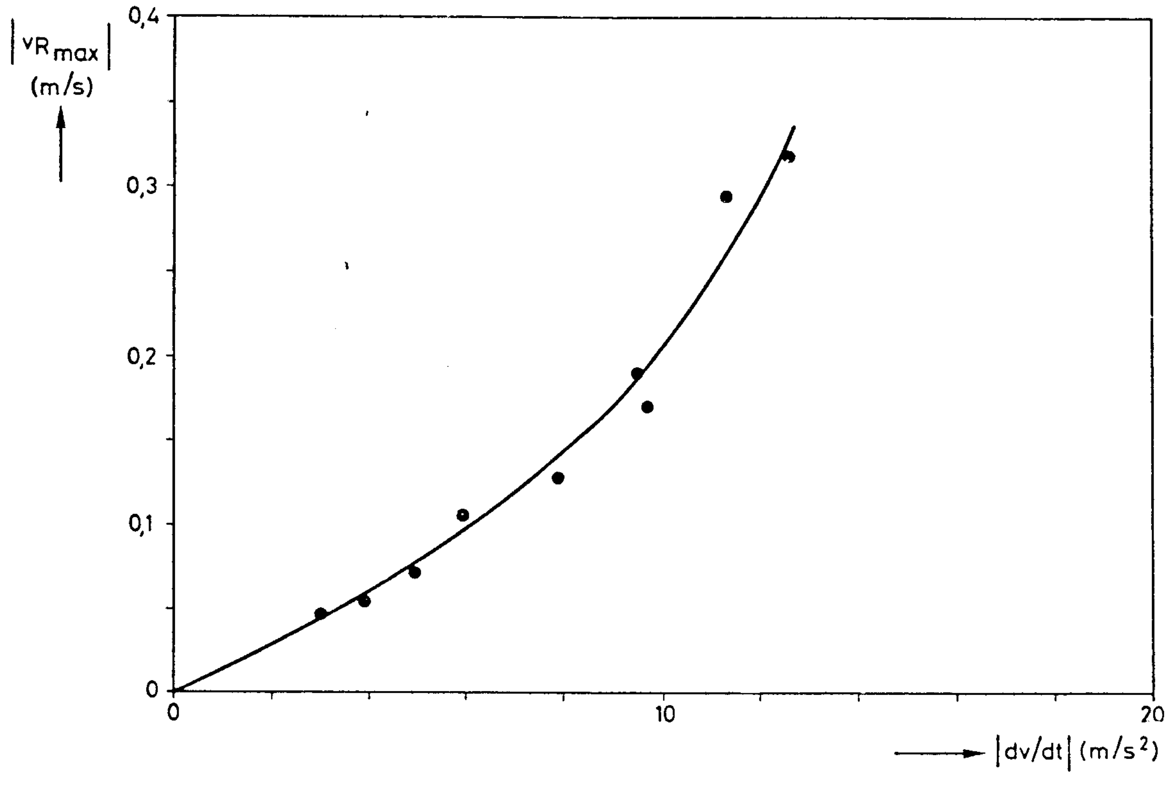

Fig. 2 Dynamic characteristic of 300 mm Mannesmann Demag Nozzle Check Valve type DRV 'B' 300/ANSI 300 with weak springs for horizontal position (valve fully open at liquid velocity of 1,24 m/s)

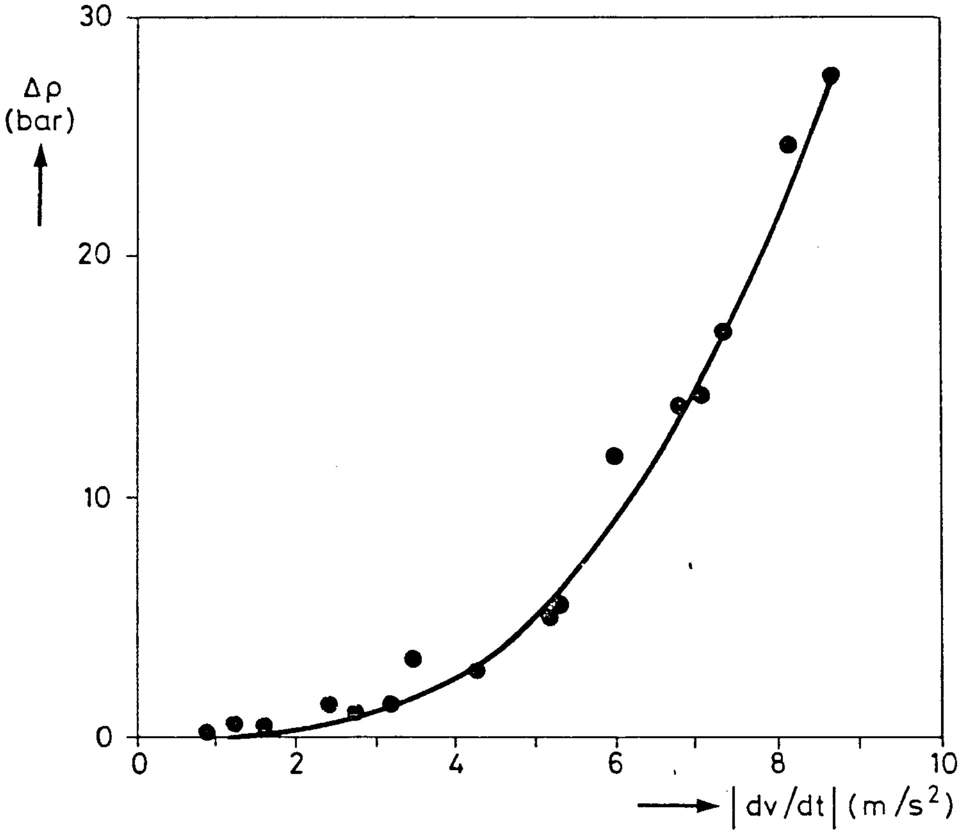

Fig. 3 Deceleration versus waterhammer pressure for 6 inch TRW Mission Dual Flapper Type Valve with improved springs DC-90 for horizontal position (after Ref. 8)

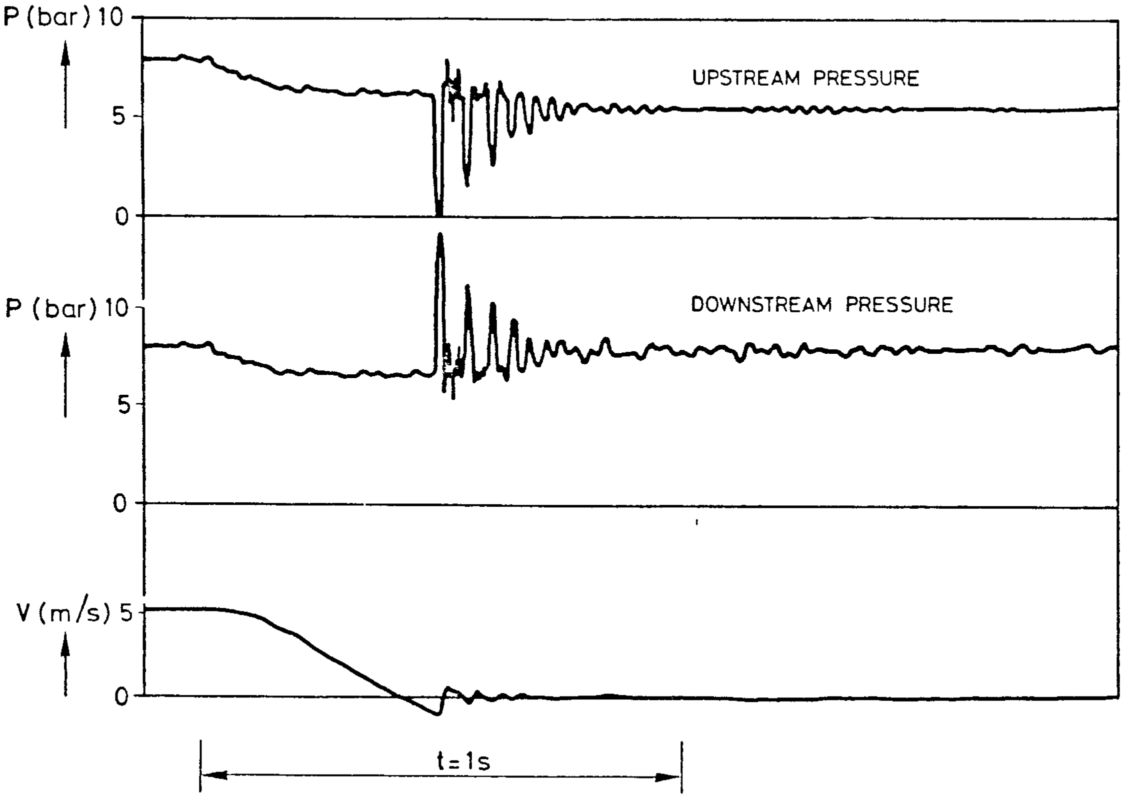

Fig. 4 Typical recording of a dynamic test of 200 mm Gestra Dual Flapper Type Valve BB16 with super strong springs

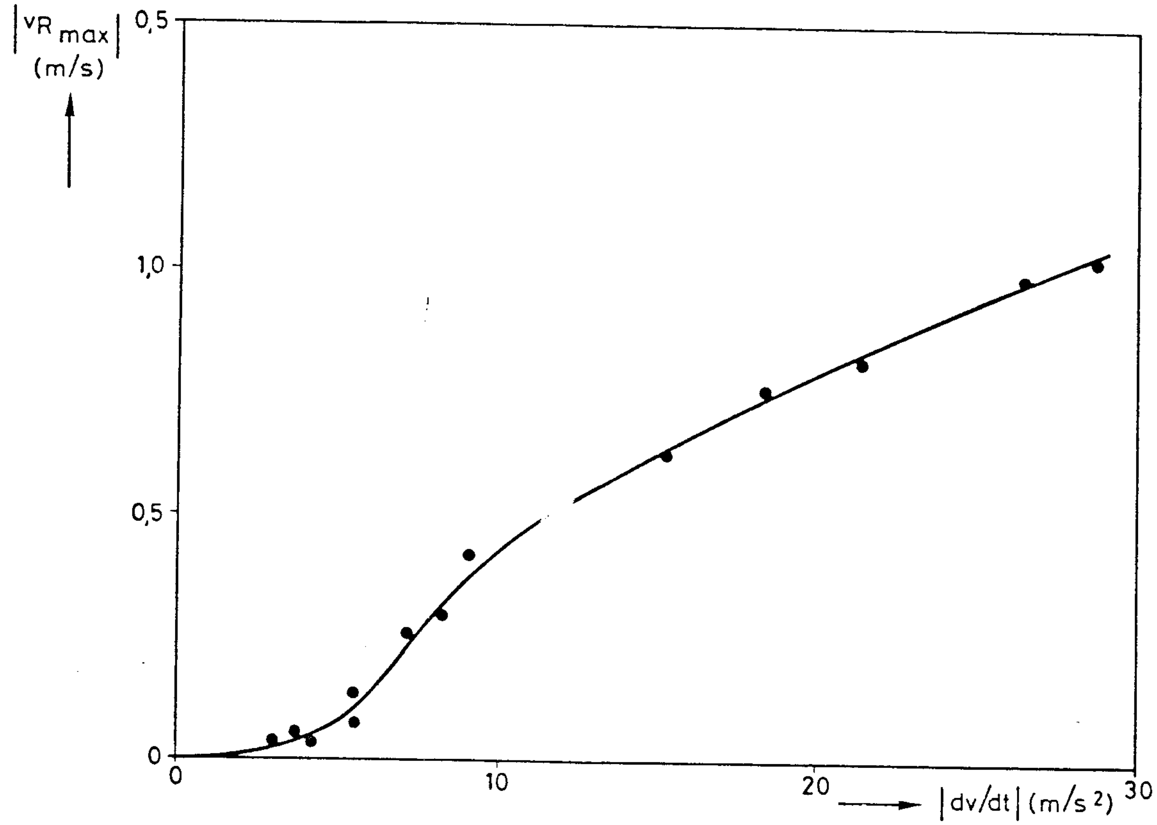

Fig. 5 Dynamic characteristic of 200 mm Gestra Dual Flapper Type Valve BB16 with super strong springs for horizontal position (valve fully open at liquid velocity of 4,8 m/s)

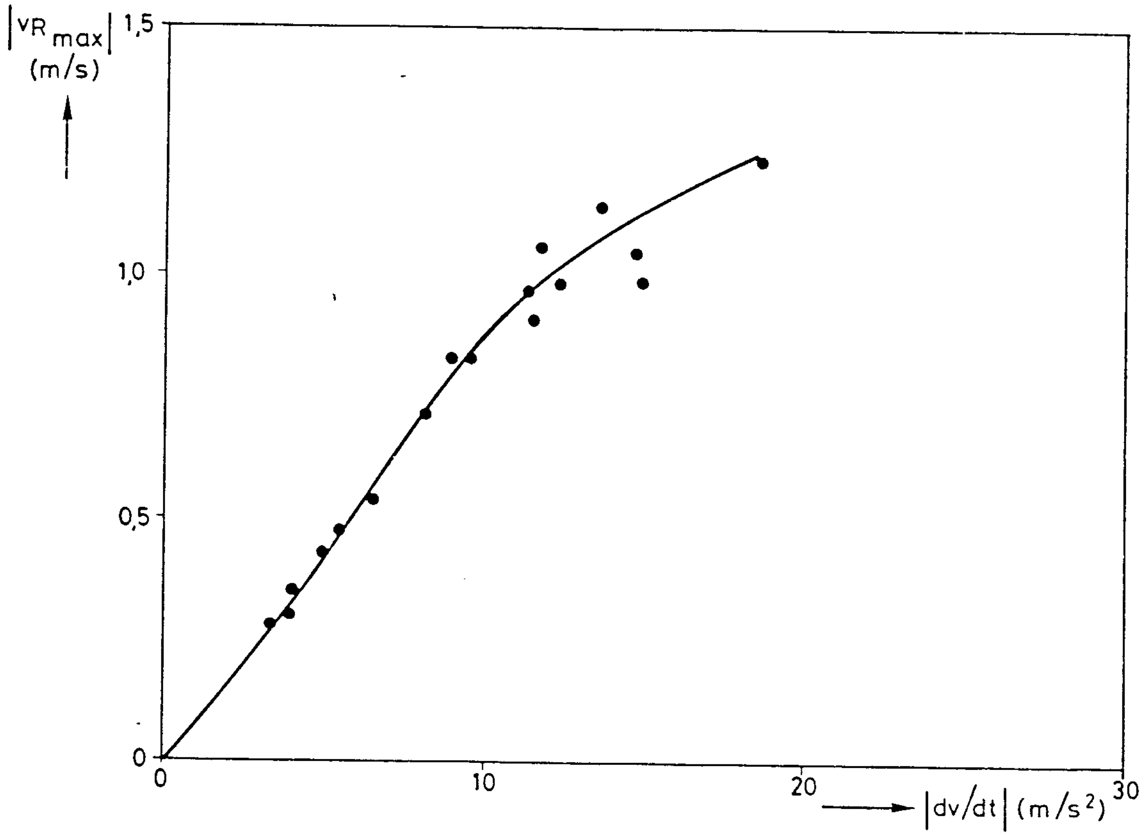

Fig. 6 Dynamic characteristic of 200 mm Gestra Translating Disc Type Valve RK56 with weak spring for horizontal position (valve fully open at liquid velocity of 0,5 m/s)

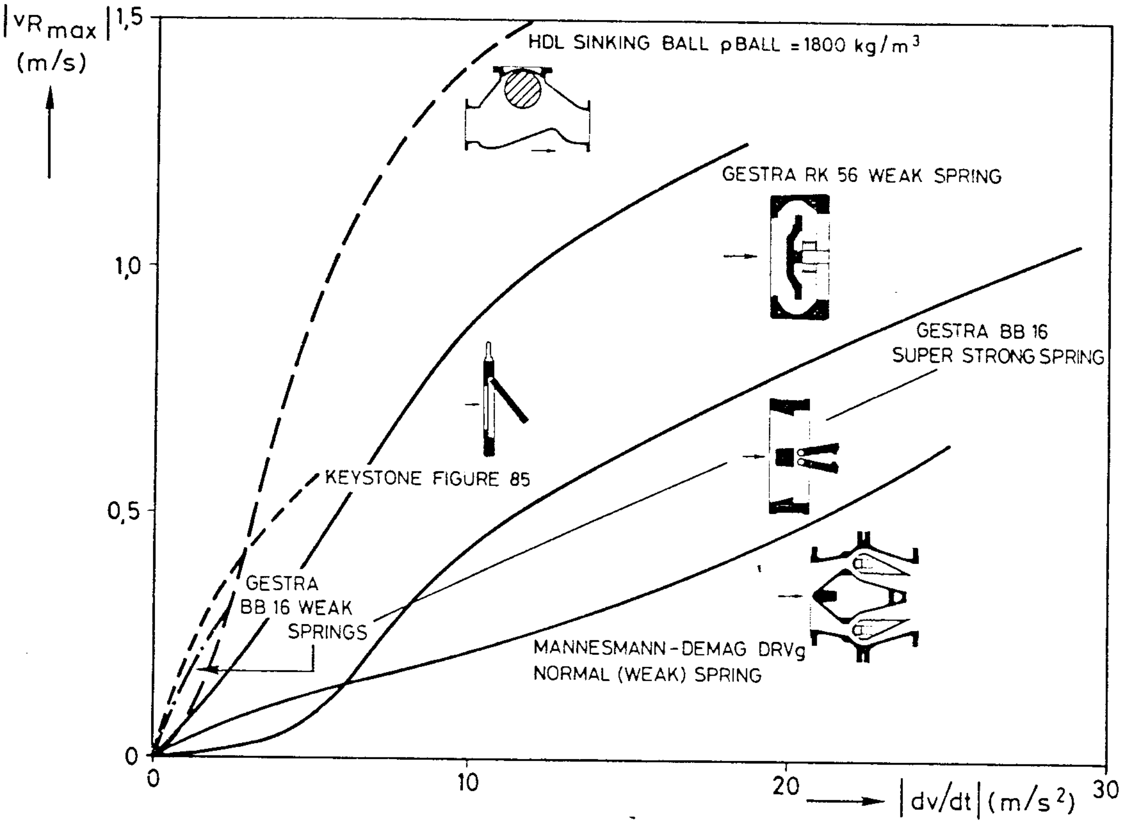

Fig. 7 Dynamic characteristics for several 200 mm non-return valves for horizontal position

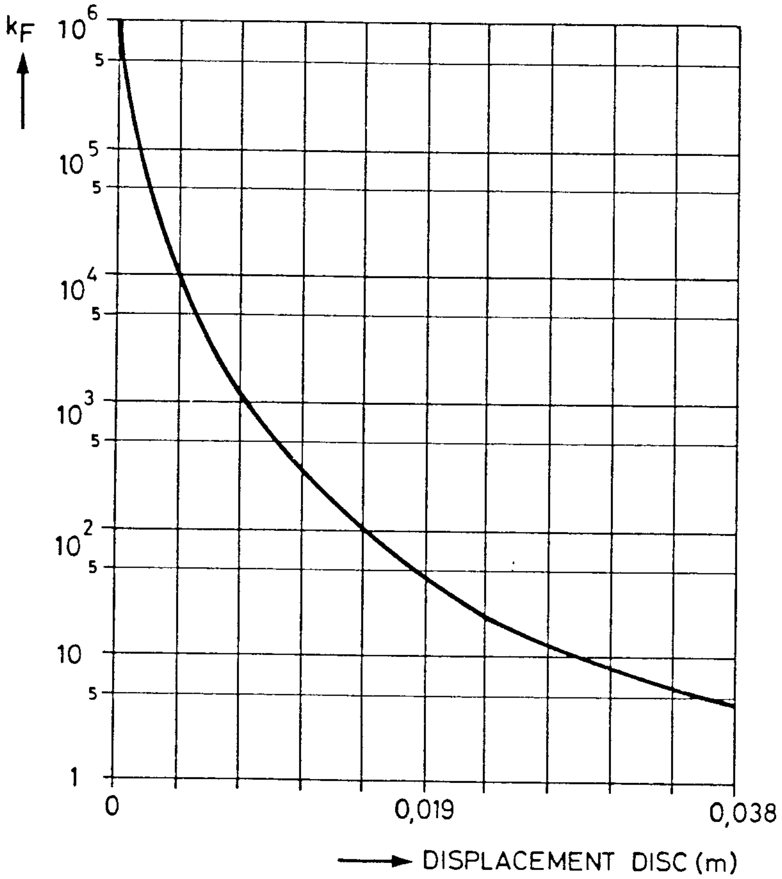

Fig. 8 Hydrodynamic Force Coefficient for 200 mm Gestra RK56 valve

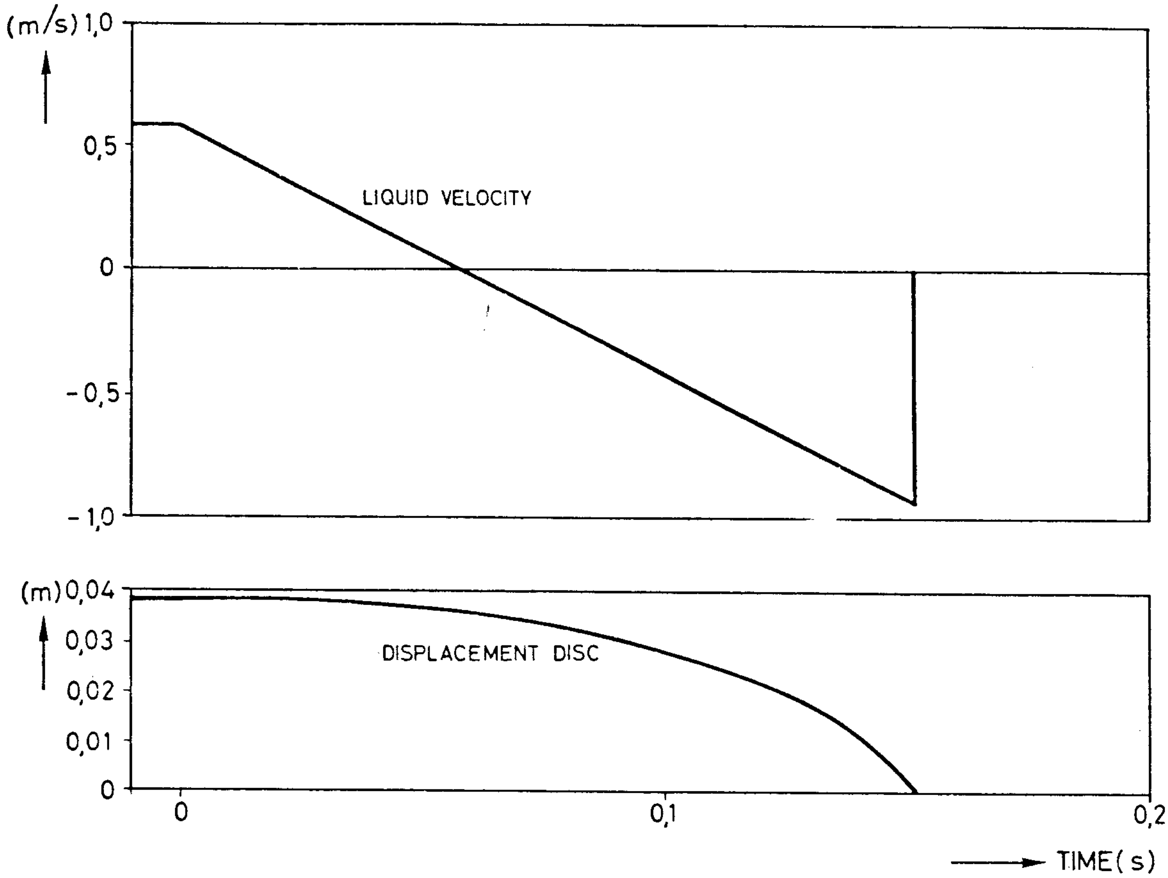

Fig. 9 Typical result of calculated liquid velocity and disc displacement versus time

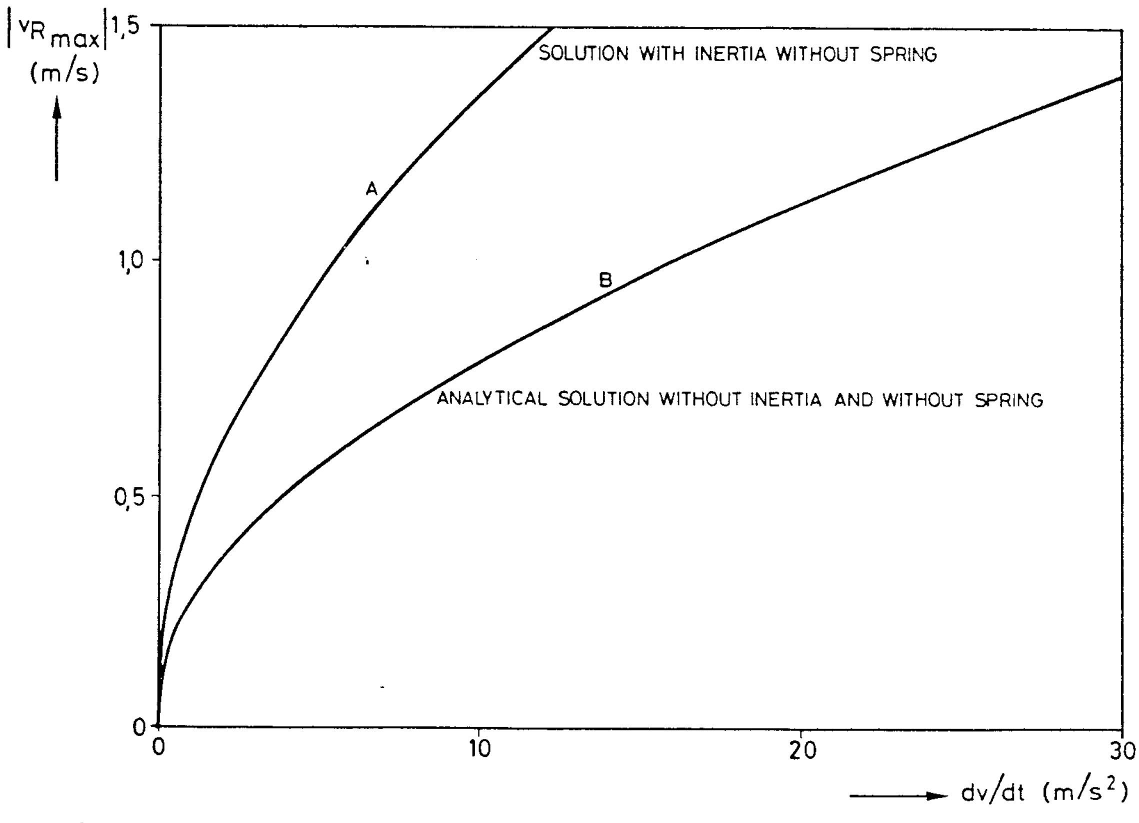

Fig. 10 Calculated dynamic characteristics for Gestra RK56 valve for horizontal position showing inertia effect

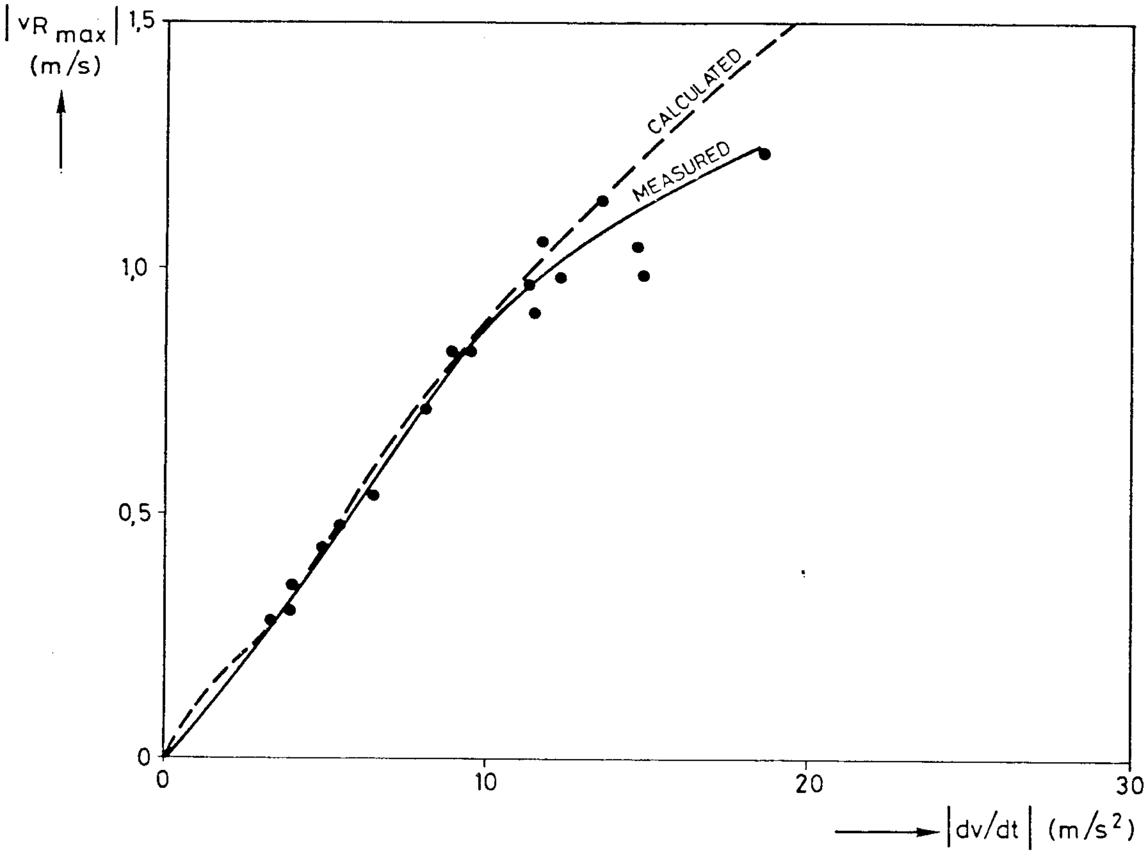

Fig. 11 Comparison between calculated and measured dynamic characteristic for Gestra RK56 valve provided with weak spring for horizontal position

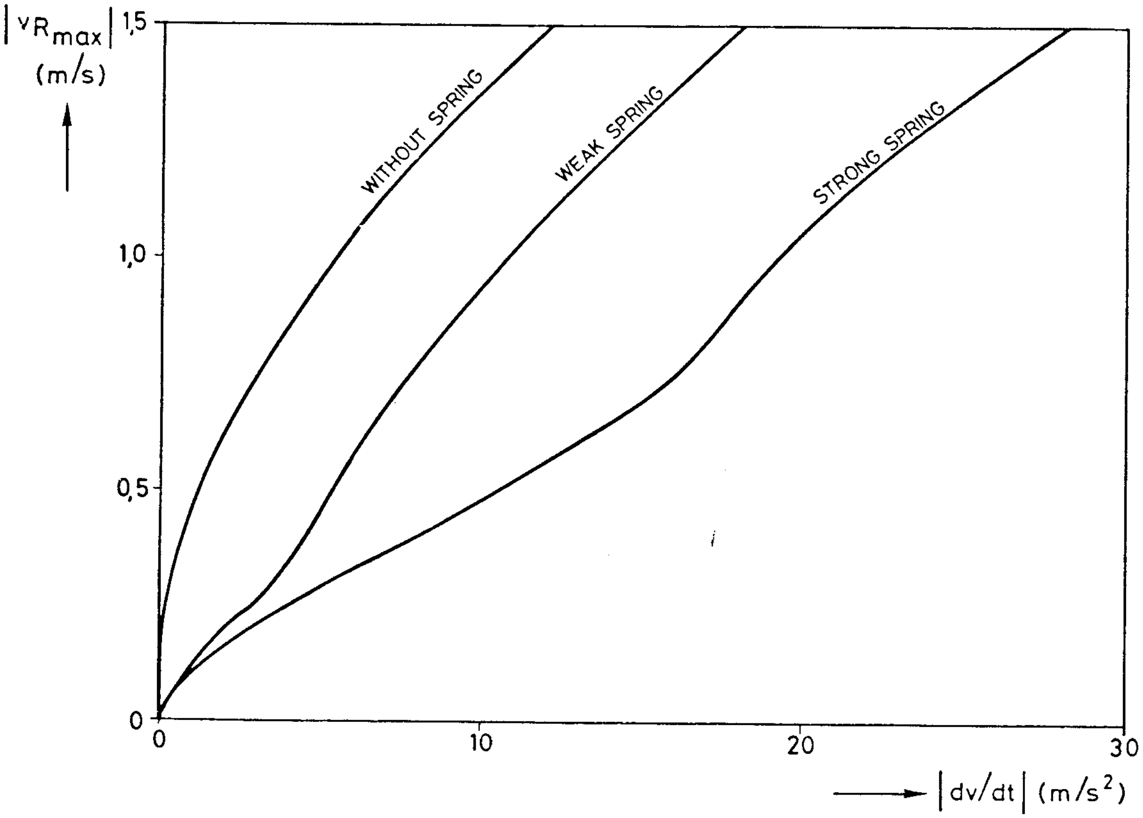

Fig. 12 Calculated dynamic characteristics for Gestra RK56 valve for horizontal position showing spring load effect

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HIGH VELOCITY OSCILLATING JET GENERATED BY WATERHAMMER (RAM-JET)

A.P. Szumowski, A.W. Laake, G.E.A. Meier, A. Unrau

The authors are Dr. A. Szumowski from the Department of Aeronautics and Applied Mechanics at the Technical University of Warsaw (Poland) and Dipl.-Phys. A.W. Laake, Dr. G.E.A. Meier and cand.phys. A. Unrau from the Max-Planck-Institut für Strömungsforschung in Göttingen (Fed. Rep. of Germany).

SYNOPSIS

An oscillating jet of high velocity liquid has been generated in a hydraulic system consisting of a tube with a buffer tank at one end and a spring loaded valve-nozzle combination at the other end. A numerical method has been developed to determine the flow properties in this system including the amplitude and the oscillation frequency assuming that the density of the liquid remains constant during the acceleration phase of the oscillation cycle and an inertialess nozzle. The theoretical predictions based on this method are confirmed by experimental results. High speed photographs are presented of the jet flow emerging from the nozzle.

NOTATION

A	bore area of the tube
a_t	sound velocity in the liquid filling an elastic tube
D	tube diameter immediately upstream of the nozzle
D_j	jet diameter
D_n	nozzle diameter
D_t	tube diameter
$k, \overline{k}$	spring constant, $k = L/Ap_o$
L	tube length
M	v_{max} / a_t
$m, \overline{m}$	nozzle mass, $m = m/ \rho AL$
p	pressure
$p_A, \overline{p}_A$	pressure immediately downstream of the tube inlet, $\overline{p}_A = p_A/p_o$
$p_B, \overline{p}_B$	pressure immediately upstream of the nozzle, $\overline{p}_B = p_B/p_o$
p_o	supply pressure
$t, \overline{t}$	time, $\overline{t} = tv_{max}/L$
$R, \overline{R}$	force exerted by the liquid on the nozzle, $R = R/Ap_o$
$u, \overline{u}$	nozzle velocity, $\overline{u} = u/v_{max}$
$v, \overline{v}$	flow velocity in the tube, $\overline{v} = v/v_{max}$
$v_B, \overline{v}_B$	flow velocity immediately upstream of the nozzle, $\overline{v}_B = v_B/v_{max}$
v_{max}	maximal flow velocity $\sqrt{(2p_o/\rho)}$
v_ϕ	flow velocity in the outlet cross-section of the nozzle
x	coordinate along the tube
$y, \overline{y}$	coordinate of the nozzle position measured from the seat, $\overline{y} = y/L$
$y_{max}, \overline{y}_{max}$	position of the nozzle stop, $\overline{y}_{max} = y_{max}/L$
$\Delta y, \Delta\overline{y}$	spring deflection, $\Delta\overline{y} = \Delta y/L$
β	$(D_t/D)^2$
φ	$(D_j/D)^2$
λ	friction coefficient
ρ	density
η	$\lambda L/D_t$

All pressures are related to the ambient pressure.

INTRODUCTION

Self-induced oscillating flow appears frequently in hydraulic lines as a result of the interaction between the flowing liquid and valves. This type of flow oscillation occurs in a hydraulic ram, known for over hundred years as a water pump [1-4]. This paper considers the ram-jet as a simple hydraulic system for the generation of a high velocity oscillating water jet, which can be used for hydraulic mining, cleaning and other purposes. It has been shown that a high speed unsteady water jet is much more effective for hydraulic mining and cleaning than a steady one [5].

The ram-jet (fig.1) consists of a buffer tank (1), a tube (2) and a spring loaded valve-nozzle (3). It operates similar to the hydraulic ram. The system is fed with water under supply pressure. When the valve-nozzle opens, the flow velocity rises and exerts a stronger stagnation pressure force on the nozzle. This force overcomes the spring resistance and the valve-nozzle moves downstream thus choking the flow. When the valve-nozzle suddenly comes to rest on the nozzle seat, a compression wave appears at the nozzle and propagates upstream. The high pressure induced by this wave produces a jet of high velocity from the mouth of the nozzle. The compression wave arriving at the buffer tank is reflected as a rarefaction wave moving downstream. This wave reduces the pressure in the tube and hence the jet velocity when it reaches the nozzle. The reduced pressure on the nozzle then enables the spring to move the nozzle away from the seat. In the following phase the flow velocity increases again due to further reflected waves. This in turn produces a higher pressure at the nozzle, the nozzle moves downstream and closes again. The oscillation cycle is thus completed and the next cycle follows.

The flow properties in the tube and thus the jet velocity depend of course on the supply pressure as well as on the dimensions of the system. The theory is based on a one-dimensional model.

GOVERNING EQUATIONS

An unsteady one-dimensional flow in an elastic tube is described by the continuity equation

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and the momentum equation

$$\frac{\partial v}{\partial t} + v\frac{\partial v}{\partial x} + \frac{1}{\rho}\frac{\partial p}{\partial x} = -\frac{\lambda}{D_t}\frac{v\,|\,v\,|}{2} \quad . \qquad (2)$$

The right hand sides of equations (1) and (2) express the influence of the elasticity of the tube and the wall friction respectively. The characteristics of equations (1) and (2) are

$$\frac{dx}{dt} = v \pm a_t \,. \qquad (3)$$

The flow velocity v in equation (3) may be neglected if the wave velocity a_t is large in comparison with v. The compatibility equations corresponding to the characteristics (3) are

$$dp \pm \rho a_t dv = \mp \frac{\lambda}{D_t}\rho\frac{v\,|\,v\,|}{2}dx \quad . \qquad (4)$$

As the variations of the density of the liquid and the flow velocity are very small, equation (4) may be partly linearized and integrated as follows

$$p \pm \rho a_t v \pm \lambda\frac{1}{D_t}\rho\int\frac{v\,|\,v\,|}{2}dx = const. \qquad (5)$$

In the case of incompressible flow, equations (1) and (2) may be integrated along the stream lines, leading to Bernoulli's equation for unsteady flow

$$p + \frac{\rho v^2}{2} + \lambda\frac{x}{D_t}\rho\frac{v\,|\,v\,|}{2} + \rho\frac{d}{dt}(xv) = const. \qquad (6)$$

BOUNDARY CONDITIONS

For constant pressure in the buffer tank the boundary condition at the buffer end is given by the stationary energy equation $p_0 = p + v^2/2$. The boundary conditions at the nozzle end are given by the modified equations for continuity

$$v_B - u = \Phi\,(v_\Phi - u) \qquad (7)$$

and energy

$$p_B + \frac{\rho}{2}(v_B - u)^2 = \frac{\rho}{2}(v_\Phi - u)^2 \quad . \qquad (8)$$

Equations (7) and (8) correspond of course to a very simplified model of the much more complicated real flow in the valve-nozzle and have been shown to be valid for an equivalent nozzle having a variable exit cross-section equal to that of the considered valve-nozzle system [6].

The relative area of the outlet cross-section depends on the nozzle position as follows

$$\Phi = \begin{cases} \phi & for \quad y = 0 \\ \phi + 4y\frac{D_n}{D^2} & for \quad y \le y_* \\ \phi + 1 - \left(\frac{D_n}{D}\right)^2 & for \quad y \ge y_* \,, \end{cases} \qquad (9)$$

where y_* is the nozzle position when

$1 - (D_n/D)^2 = 4y\,D_n/D^2$.

The nozzle travel typically is limited by a stop. The position of the stop y_{max} determines the maximum value of Φ corresponding to $y_{max} < y_*$. In the opposite case Φ_{max} is equal to the value given by the last of equations (9). The nozzle position in the range $0 < y < y_{max}$ is determined by the equation of motion for the valve-nozzle

$$-m\frac{d^2y}{dt^2} = R - k\,(\Delta y - y) \quad , \qquad (10)$$

in which the friction term is neglected. The force R is produced by the flow pattern just upstream of the valve-nozzle, which strongly depends on the valve-nozzle position. The following approximate expression for R can may be introduced

$$R = A_n\left[p_B + \rho(v_B - u)^2\right] - \frac{A_j}{\Phi^2}\rho(v_B - u)^2 . \qquad (11)$$

Expression (11) results from continuity and momentum equations for the control surface marked in figure 4 by broken lines. In the momentum equation the momentum exchange through the side surface of the control volume has been neglected.

MODELS OF FLOW PATTERN

The wave pattern described in the introduction may be illustrated in the x,t-plane as shown in figure 2a. The continous and broken lines in this figure indicate the trajectories of compression and rarefaction waves respectively. Only the main waves are shown. The secondary waves, which appear as a result of the tube motion are neglected. The oscillation period T is the sum of the time intervals ΔT_1 and ΔT_2. ΔT_1 corresponds to the jet phase where the valve-nozzle rests on the seat and hence the jet velocity is high. ΔT_2 corresponds to the acceleration phase of the water column in the tube where the jet velocity is low and the flow velocity grows from nearly zero to the value which is necessary to move the nozzle in the direction of the seat. We assume that the flow in the tube is directed to the valve nozzle in each phase of the oscillation cycle. A more general case taking into account also opposite flow direction is considered in [7].

In addition to the wave pattern the phase diagram of pressure and velocity may also be used to illustrate the transients of the pressure waves (fig.3). Each point in the phase diagram corresponds to the conditions behind a single wave front. The numbers of the different points in the phase diagram correspond to the states behind the like numbered pressure waves in the wave pattern (fig.2a). The transitions between the different states of pressure and velocity are determined by the compability equations (4) and the boundary conditions at both ends of the tube. In the phase diagram these transitions are illustrated by thin continous lines. The boundary conditions at the buffer tank and at the valve nozzle are shown by thick continuous and dashed lines respectively.

INCOMPRESSIBLE FLOW DURING THE ACCELERATION PHASE

The waves appearing in the acceleration phase ΔT_2 (thin lines in fig.2) are weak in comparison with the primary compression and rarefaction wave (thick lines). Therefore, a considerable number of these weak waves is necessary to accelerate the liquid. Thus the acceleration-phase ΔT_2 must be long in comparison to the jet-phase ΔT_1. For the weak waves of the acceleration phase the incompressible flow during the ΔT_2 may be assumed. In this case the wave diagramm can be simplified as shown in figure 2b. The cycle begins at the time t_1 where the valve-nozzle rests on the seat. During the following acceleration phase the valve-nozzle moves from the seat to the stop and back. When the valve-nozzle arrives at the seat a compresion wave ocurs and is reflected from the buffer tank. The initial condition for the flow in the acceleration phase is determined by the flow properties just behind the reflected rarefaction wave. In the case considered in this paper the flow velocity behind the rarefaction wave is directed into the buffer tank. Thus during ΔT_2 the following energy equation must be satisfied

$$p_0 = p_B + \frac{\rho v^2}{2} + \rho L \frac{dv}{dt} + \lambda \frac{L}{D_t} \frac{\rho v^2}{2} \quad . \tag{12}$$

Introducing dimensionless parameters and boundary conditions at the nozzle end, we get from equations (7,8,10-12) the following expressions for the derivative of flow and nozzle velocities and the nozzle position

$$\frac{d\bar{v}_B}{d\bar{t}} = \frac{\beta}{2}\left[1 - \bar{p}_B - \frac{\bar{v}_B^2}{\beta^2}\left(1 + \lambda \frac{L}{D_t}\right)\right] \quad , \tag{13}$$

$$\frac{d\bar{u}}{d\bar{t}} = \frac{1}{\bar{m}}\left[\bar{R} - \bar{k}\left(\Delta\bar{y} - \bar{y}\right)\right] \quad , \tag{14}$$

$$\frac{d\bar{y}}{d\bar{t}} = -\bar{u} \quad , \tag{15}$$

where

$$\bar{p}_B = \left(\bar{v}_B - \bar{u}\right)^2 \left(\tfrac{1}{\Phi^2} - 1\right)$$

and

$$\bar{R} = \left(\bar{v}_B - \bar{v}\right)^2 \left[\left(\tfrac{D_n}{D}\right)^2 - \left(\tfrac{1}{\Phi^2} + 1\right) - \tfrac{2\phi}{\Phi^2}\right] \quad ,$$

$$\bar{v}_B = \bar{v}\beta \quad .$$

Using equations (13-15) the flow properties at the time t_2 (fig.2b) can be calculated. On the other hand the flow properties at the time instants t_1 and t_2 are coupled by compability equations for compression and reflected rarefraction waves as well as by the boundary condition at the nozzle end. During the jet-phase ΔT_1 the flow velocity in the tube is relatively small. In this case the friction term in equation (5) can be neglected. Under this assumption we have from equations (5,7 and 8) for the compression wave

$$\bar{v}_{B2} = \frac{1}{\left(\frac{1}{\phi^2} - 1\right)}\left[-\frac{1}{\beta M} + \sqrt{\frac{1}{(\beta M)^2} + \left(\frac{1}{\phi^2} - 1\right)\left(\bar{p}_{B1} + \frac{2}{\beta M}\bar{v}_{B1}\right)}\,\right] , \tag{16}$$

$$\bar{p}_{B2} = \bar{p}_{B1} + \frac{2}{\beta M}\left(\bar{v}_{B1} - \bar{v}_{B2}\right) \tag{17}$$

and for the rarefraction wave

$$\bar{v}_{B3} = \frac{1}{\left(\frac{1}{\phi^2} - 1\right)}\left[\frac{1}{\beta M} - \sqrt{\frac{1}{(\beta M)^2} + \left(\frac{1}{\phi^2} - 1\right)\left(\bar{p}_{B2} - \frac{2}{\beta M}\bar{v}_{B2}\right)}\,\right] , \tag{18}$$

$$\bar{p}_{B3} = \bar{p}_{B2} - \frac{2}{\beta M}\left(\bar{v}_{B2} - \bar{v}_{B3}\right) \quad . \tag{19}$$

CASE OF THE INERTIALESS VALVE-NOZZLE

Normally, the mass of the valve-nozzle is much smaller than the mass of the water column in the tube. In this case the mass of the nozzle can be neglected. Using the model of incompressible flow in the acceleration phase ΔT_2 this assumption and the assumption of a constant spring force during the nozzle motion equal to its value at y=0 (which is admissible when the nozzle travel y_{max} is sufficiently small in relation to the spring deflection Δy), allow to solve equation (13) analitically.

$$\bar{v}_B = \beta \frac{r}{q} \frac{e^{rq(\bar{t} - \bar{t}_1)} - \varepsilon}{e^{rq(\bar{t} - \bar{t}_1)} + \varepsilon} \quad , \tag{20}$$

where

$$r = \begin{cases} \sqrt{1 - \bar{k}\Delta\bar{y}/\left[\left(\frac{D_n}{D}\right)^2 - 2\phi\right]} & \text{the valve-nozzle moves} \\ 1 & \text{the valve-nozzle rests on the stop} \end{cases}$$

$$q = \begin{cases} \sqrt{1 + \eta} & \text{the valve-nozzle moves} \\ \sqrt{1 + \eta + \left(\beta/\Phi_{max}\right)^2} & \text{the valve-nozzle rests on the stop} \end{cases}$$

$$\varepsilon = \frac{r - q\bar{v}_{B3}/\beta}{r + q\bar{v}_{B3}/\beta} \quad .$$

From this equation the duration of the acceleration phase ΔT_2 can be obtained. In this way one obtains the oscillation frequency of the ram-jet.

EXPERIMENT

Experiments were conducted in an apparatus employing a steel tube of 1 1/4" diameter and 0.95 m length. The buffer tank consists of a short tube of 2" diameter filled with cellular rubber. The valve-nozzle section shown in figure 5 has a moving mass of m=0.042 kg returned by a coil spring of k=7321 N/m. The entire system was suspended with rubber shock cords. A pump with a regulated outlet pressure was used to supply of water to the system. Both the supply pressure p and the nozzle travel y_{max} were controlled during the experiments. For given p and y_{max}, the pressure transients at both tube ends as well as the acceleration of the tube itself were measured. Kistler pressure transducers (types 601A and 6031) and a Kistler accelerometer (type 816A) were used. Additionally high speed photographs of the jets were taken at several times during the oscillation cycle.

RESULTS AND DISCUSSION

An example of measured pressure and acceleration traces is shown in figure 6. The pressure signal from the transducer placed close to the nozzle shows regular peaks, induced by waterhammer, appearing in the time intervals ΔT_1, while the valve-nozzle rests on the seat. The pressure traces at both tube ends are seen to differ from one another. As a result, an unbalanced longitudinal force appears which involves the tube motion as indicated by the accelerometer records. The signal from the accelerometer shows high frequency tube vibrations resulting from the transformation of pressure energy to acceleration of the tube.

Using the model described above the values of pressure and velocity have been calculated corresponding to the experimental parameters. These results are shown in figure 7. In these calculations a fixed tube and a constant pressure at the buffer end were assumed. The measured and the calculated oscillation frequency as well as the pressure amplitude are compared in figure 8. Since the agreement between the values calculated those measured is satisfactory, it can be concluded that the inertia of the valve-nozzle has only a minor influence on the flow within the range of parameters considered. It appears then, that the assumption of an inertialess valve-nozzle leads to an adequate model of the flow considered.

In general, the oscillation frequency grows as the duration of the acceleration phase decreases. An increase in the spring force, the nozzle travel or the relative outlet cross-section causes an increase in the acceleration time interval ΔT_2. In the same way a decrease of the cross-section ratio or of the supply pressure leads to an increase in the acceleration time interval ΔT_2 and thus to a decrease of the oscillation frequency. Simultaneously, longer ΔT_2 lead to a higher flow velocity at the moment, when the valve-nozzle stops on the seat. This in turn increases the pressure amplitude. The high speed photographs show three states of the oscillation cycle with a supply pressure of 0.127 MPa and a nozzle travel of 9.3 mm. Figure 9a shows the initial state of the flow through the open nozzle, figure 9b corresponds to the acceleration phase. Figure 9c shows the high pressure jet flow generated when the nozzle stops on its seat.

CONCLUSIONS

The hydraulic system considered here can be effectively exploited to generate an oscilllating jet of high velocity and pressure. The velocity and pressure amplitudes strongly depend on the system dimensions and of course on the supply pressure. The oscillating flow appears accurately predicted analytically using the model which incorporates an assumption of incompressible liquid flow during the acceleration phase and an inertialess nozzle. The theoretical and experimental results show satisfactory agreement, in spite of some minor discrepancy between the theoretical model and detailed experimental observations. It appears, that the model can be used effectively to design practical systems.

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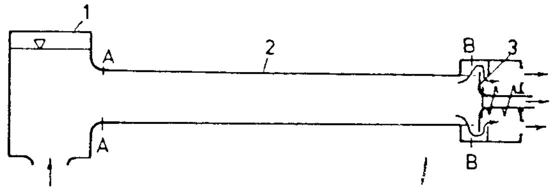


Fig.1: Schematic diagram of the ram-jet
1 - buffer tank, 2 - tube, 3 - elastically
supported convergent valve-nozzle

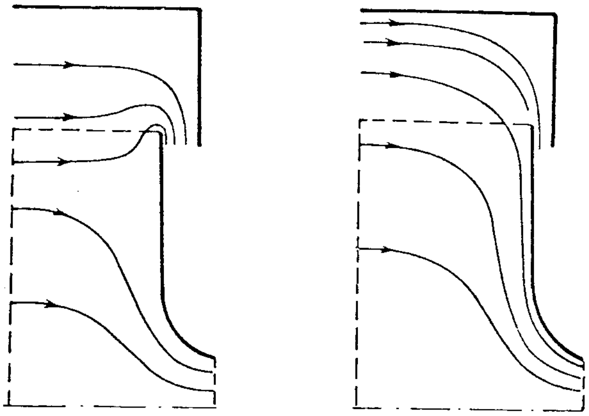


Fig.4: Flow pattern in the nozzle section for
a large y (a) and a small y (b)

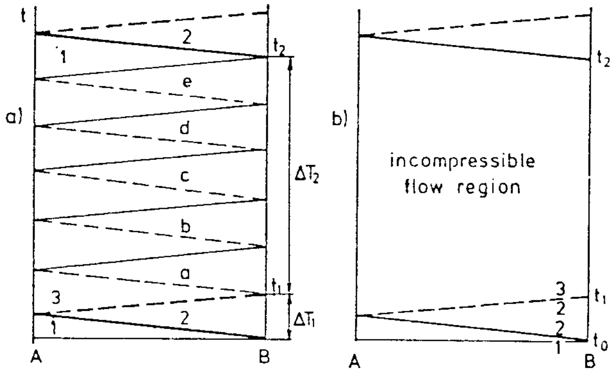


Fig.2: Wave pattern diagram

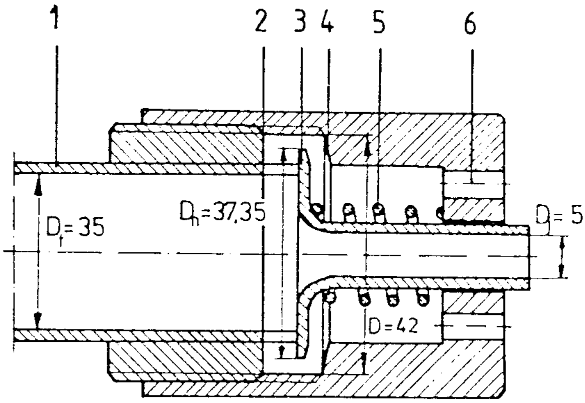


Fig.5: Valve-nozzle section of the ram-jet
1 - tube, 2 - nozzle stop, 3 - valve-nozzle,
4 - nozzle seat, 5 - spring, 6 - outlet

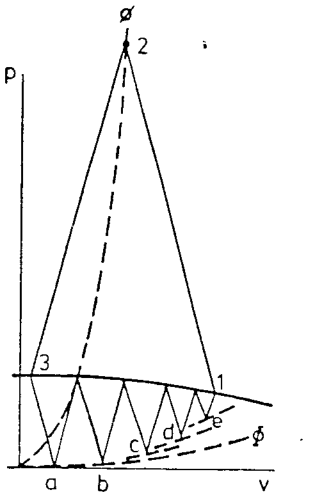


Fig.3: Phase diagram

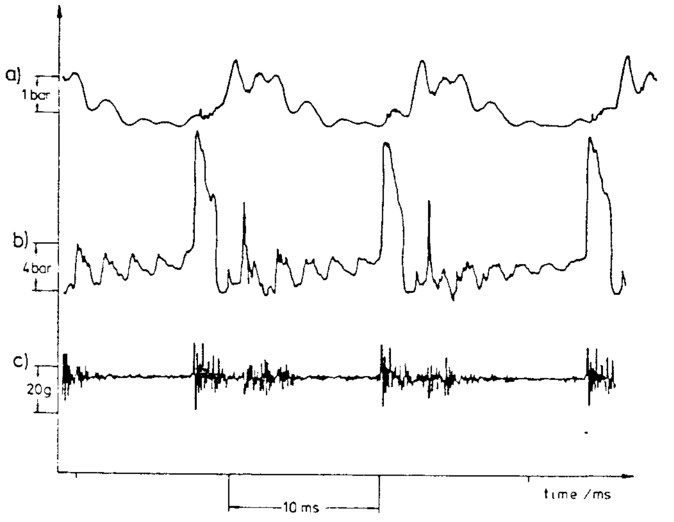


Fig.6: Measured pressure oscillations at the buffer tank (a) and at the nozzle (b) compared with the tube acceleration (c) $p_o = 2.2$ bar, $\Delta y = 13$ mm, $y_{max} = 1$ mm

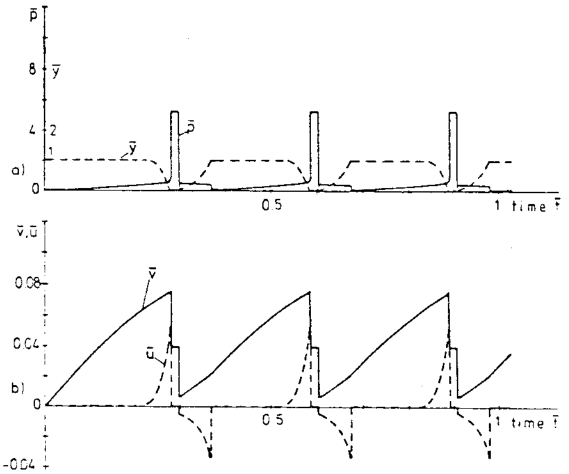


Fig.7: Calculated traces

a)—— pressure at the nozzle ---- nozzle position

b)—— flow velocity ---- nozzle velocity

p_0 = 2.2 bar, Δy = 13 mm, y_{max} = 1 mm, λ = 0.03

a_f = 1150 m/s, $\bar{m}$ = 0.0319

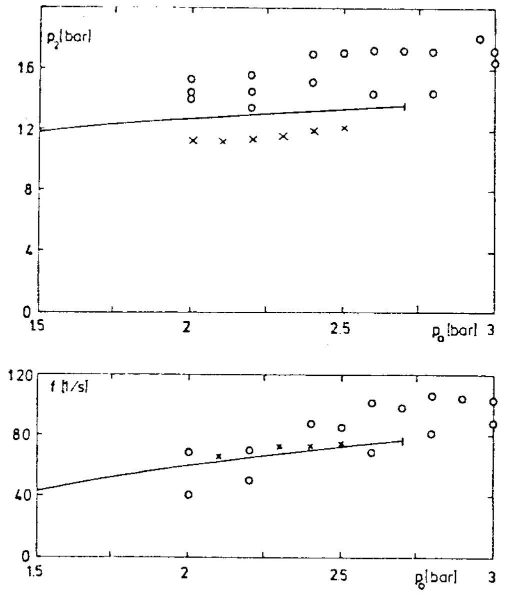
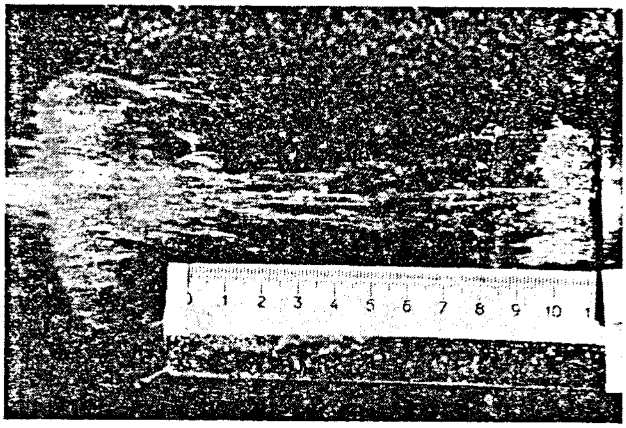


Fig.8: Pressure amplitude and oscillation frequency versus supply pressure

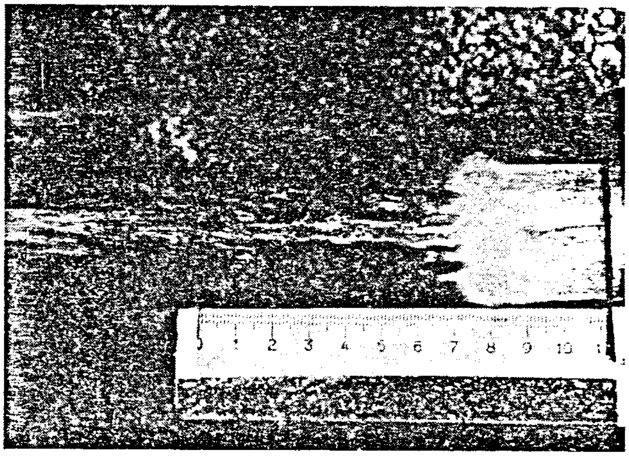
---- analytical method (inertialess nozzle)

x numerical method

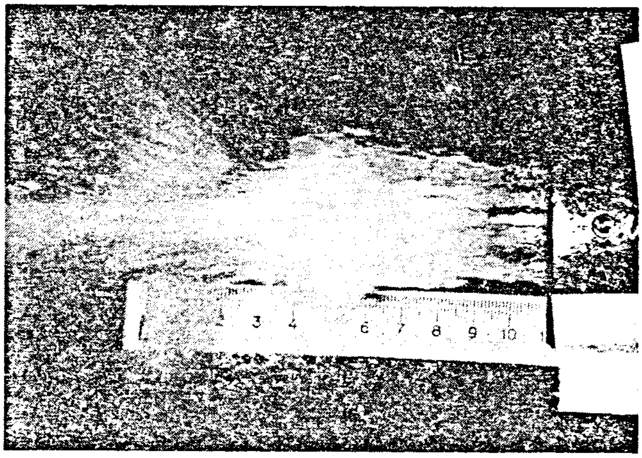
o experiment (experimental data in fig. 7)



(a) initial state



(b) acceleration phase



(c) jet phase

Fig.9: High speed photographs of jet

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PAPER H3

p. 229-235

CHECK VALVE DYNAMICS IN PRESSURE TRANSIENT ANALYSIS

H.-D. Perko

Mannesmann Demag Meer
Check Valve Department
Ohlerkirchweg 66
D-4050 Mönchengladbach 1

Summary

The dynamic behaviour of check valves can be an important factor in the design of pipe systems. In this paper check valve closure and the resultant pressure fluctuations are gone into in some detail. Results of field measurements as well as numerical simulations are presented.

Criteria for aiding the valve selection, such as displacement volume and the eigenfrequency of the moving parts and the natural closing time of the valve are discussed.

Reference is also made in the paper to how in the case of cavitation the subsequent water hammer can be prevented with a check valve in the line.

Introduction

In most pipeline systems check valves are essential components. Their function is to limit backward flow in respect of reverse rotation of pumps and emptying of reservoirs or pipelines. Under transient conditions check valves can also contribute actively to causing pressure surges.

Badly selected check valves can even be a danger to the plant. The pressure can exceed the allowable limit and the valves can even be severely damaged when they slam too hard. Mostly, the two occur in combination with each other, but in short pipes slamming of the valves is not necessarily connected with a high pressure surge.

In recent years there has been an increased effort to take into account check valves in the pressure transient analysis. For this purpose reliable information on dynamic properties must be available. Test rigs have been developed for investigating the dynamic behaviour of different types of check valves.

Principle of Check Valve Closure

To analyse the slamming phenomenon one can limit the consideration to the pipe connected to the check valve. Under transient conditions, e.g., after a pump stop, the flow through the valve starts to decelerate. When the flow velocity has reached a valve smaller than the fully open velocity of the valve v_0, the valve starts to close. Owing to the inertia and the travel distance of the moving parts, the valve closes after backflow is reached (Fig. 1). We shall refer to the maximum return velocity as v_r.

Undampened valves close normally in such a way that the flow changes from v_r to zero rapidly in terms of the pipeline period. This change generates a sudden pressure rise at the normal outlet side of the valve, which can be calculated from Joukowsky's well-known equation:

$$\Delta p = \rho \; c \; v_r \qquad (1)$$

where c is the propagation speed of pressure waves and ρ the density of the fluid.

In principle, the dampened valves act in the same way. Seen in Fig. 1 is a qualitative comparison of a dampened and an undampened valve of similar type. The maximum backflow of the dampened type is much higher and the time T_s for the change from v_r to zero is appreciable longer. Both depend on the time scale of the dampening mechanism.

The extent of the pressure rise depends as a first approximation on the amount of v_r and the ratio between T_s and the travel time T_r of a pressure wave to the next reflection point of the pipe system.
If T_s is smaller than 2 T_r, then an even higher pressure rise is generated compared with an undampened valve. A reduction in the pressure rise can only be obtained when the dampening time T_d is chosen in such a way that T_s is a multiple of 2 T_r.
The longer T_d is the smaller is the pressure rise caused by the check valve. On the other hand the total backflow increases with an increase of T_d. Above a certain point this can mean that the check valve is not fulfilling its function of protecting the pump against reverse rotation.

If wishing to install dampened check valves one has to analyse very carefully with a pressure transient calculation the piping system, taking into account the exact technical data of the pumps and check valves to achieve a suitable value for T_d.

For a given valve, the higher the deceleration of the flow, the higher is the maximum backflow v_r and consequently the pressure rise. As a first approximation the rate of the flow deceleration can be calculated from the difference between the pressure heads at the ends of the connected pipe and the pipe length.
Shown in Fig. 2 are some typical layouts where a high flow deceleration can be expected. Layouts (a) to (c) show situations where flow deceleration is due to pump failure. Layout (d)

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shows a series pumping station. Here the flow through the check valve stops when the pump starts.

In systems similar to these shown it is necessary to predict the dynamic behaviour of the non-return valve and resultant pressure and/or to choose a quick-closing check valve.

Field Measurements

Reported in /9/ are some tests comparing a swing check valve and a Mannesmann Demag ring (annular) nozzle check valve, both having a nominal diameter of 300 mm. The tests were performed in a system with a surge vessel and with two pumps working in parallel. The total pipe length is about 6 km and the difference in water level between the two reservoirs is 47 m.

When one pump is running the head is 50 m and the delivery is 162 l/s (2.3 m/s velocity in the pipe connected to the valve). Both pumps running gives a head of 61 m and a delivery of 255 l/s (1.8 m/s velocity).

In both cases a pump failure was simulated and the pressure was recorded immediately downstream of the valves. A set of pressure-time histories is shown in Fig. 3 (retraced for clarity).

The ring nozzle type check valves show a considerably better performance under such transient conditions. All the swing check valves in this system were replaced.

In some publications /5/6/ other experimental data are presented which show similar situations where it was necessary to install check valves with very short closure times.

Theory

The motion of the moving parts of the valve is described by Newton's Second Law of Motion:

$$F_I + F_w + F_B + F_s + F_f + F_h = 0 \qquad (2)$$

The terms in this equation represent the forces exerted on the translational moving part (or torques for valve types with rotating parts) due to, respectively, inertia (of the parts and the surrounding water), gravitation, buoyancy, external forces (weight or spring forces), friction, and hydrodynamics.

A general solution of equation (2) covering different valve types cannot be found. The gravitation, buoyancy and external forces are easily determined from valve dimensions, material properties and spring forces, but the determination of the hydrodynamic and the friction forces is afflicted with great uncertainties.

Satisfactory assumptions can be found only in combination with measurements. Some acceptable solutions have been published. They are valid only for certain types of check valves, but it has been found to be sufficiently accurate to assume the hydrodynamic forces to be proportional to the square of the net flow rate passing the moving parts and to a coefficient which is solely dependent on the position of the moving parts.

A numerical solution for the ring nozzle type of Mannesmann Demag has been developed by the use of this assumption and measured values of the hydrodynamic coefficient. The numerical model is implemented in a code for the calculation of pressure transients which is based on the method of characteristics.

Results of calculations on a parallel pump station are presented in Fig. 4 and 5. Fig. 4 relates to a simulation of a ring nozzle check valve with weak springs as it is shown below in Fig. 6. The dynamic properties of a conventional swing type check valve are simulated in calculations shown in Fig. 5.

The time histories of the relative valve opening position (the dotted line represents the actual equilibrium position), the flow velocity through the valve and the pressure downstream of the valve are shown.

The calculations in Fig. 5 show a hard slamming followed by repeated opening and closing of the valve. Due to the high pressure fluctuations, cavity is formed and the pressure rise after the second closure is even higher than that after the first. The results make clear the importance of a quick-closing valve in such a piping system.

Comparative Experimental Investigations

Since an analytical method comparing the dynamic properties of different check valve types and sizes is not possible, in recent years attention has been focussed on experimental studies. In the more sophisticated tests a constant deceleration is generated and the maximum backflow velocity v_r is determined /1/2/3/6/8/.

The most published measurements were performed in the Delft Hydraulic Laboratories. In the publications by Provoost /3/ and Thorley /6/ the different dynamic behaviours of the valve types become clear. The moving ball type and the swing type give a comparatively poor response with relatively high reverse flows at modest decelerations. The split disc type is rather better, particularly with strong springs. Nevertheless, all results indicate that the ring nozzle valves give the best performance even at the highest deceleration rates.

In accordance with Provoost /2/, the dynamic characteristic shall be defined here as a graph of the velocity v_r versus the flow deceleration dv/dt.

Such a dynamic characteristic can only be applied direct in pressure transient analysis where a constant deceleration occurs similar to that in the tests, and where the check valve is fully opened in steady state. Re-opening of valves cannot be simulated.

However, the results of the measurements are very helpful for assessing the accuracy of a numerical model. Calculated and measured dynamic characteristics of tested ring nozzle type check valves are shown in Fig. 6. The solid lines represent calculations based on a solution of equation (2).

Discussion of Comparative Parameters

Up to now the most information about dynamic properties of a check valve is given with a measured dynamic characteristic. Since it would be too expensive to test each valve, criteria for the estimation of the dynamic behaviour have to be found additionally. Some ideas are outlined in the following.

If friction, external load and inertia are neglected, the dynamic characteristic can be determined analytically. The disc will follow the motion of the fluid, thus the volume of the liquid which has passed through the valve after reversal of flow will be equal to the displacement of the disc.
This volume V_D is known from the valve geometry. Compared in Fig. 7 is a so-determined dynamic characteristic of the Mannesmann Demag check valve DRV-B 300 (V_D = 1.25 dm^3) with results of calculations based on different spring loads. The spring force is expressed as the fully opening velocity v_o of valves in the horizontal position.

It becomes clear that for such a spring-loaded type of check valve (also weight-loaded swing type), taking into account only displacement volume is not sufficient when comparing different check valves. The displacement volume only gives a first approximation of which dynamic behaviour of a certain check valve can be expected; the smaller the displacement volume, the better is the dynamic characteristic.

Another important parameter is the eigenfrequency of the moving parts. Nearly all types of check valves can be regarded as a pendulum. In general it can be said that the higher the eigenfrequency the better is the dynamic behaviour. The limitation of this relation is again found in its application to weight- or spring-loaded types of check valve.
An external weight decreases the eigenfrequency of a swing type check valve, but it has the effect of, particularly at small rates of flow deceleration, a better dynamic behaviour.
For spring-loaded types the eigenfrequency is determined by inertia and spring rate. Depending on the installation position and on the amount of the spring forces, valves having the same eigenfrequency can exhibit a different dynamic behaviour.

To take these effects into consideration one can switch over to an easily determinable parameter which is connected with the eigenfrequency, namely, the time t_n which a valve needs to change from fully open to the closed position in a still fluid. We shall refer to this time as the natural closing time.
For the tested Mannesmann Demag valves (shown in Fig. 6) t_n can be given as follows (in air):

Type DRVg	200		- 29 ms,
Type DRV-B	300	with weak springs	- 28 ms,
Type DRV-B	300	with strong springs	- 10 ms.

From the measured dynamic characteristics for each rate of flow deceleration it is possible to determine the time t_r between flow reversal and valve closure. Relating t_r to t_n gives a average ratio of 1 : 1.5 within a range from 1 : 1 to 1 : 2.

In /4/ are reported some preliminary tests on a swing type check valve of 150 mm nominal size.
A natural closing time of 220 ms through an angle of 57° in air was measured. This time can be seen to be in accordance with the above relation when compared with the measurements of dynamic characteristics in /1/5/ (in spite of all uncertainties seeing that it was not the same valve in both cases).

The results of tests already performed and future measurements should be used to examine whether or not this or a similar relation can be found to be generally valid.

In these considerations concerning an easily determinable parameter for an estimation of check valve behaviour, friction forces are only included to a limited extent. Many types of check valve have to be inspected and lubricated at certain intervals, otherwise after some years in use the friction forces can increase significantly due to corrosion and other contamination of surfaces. This can have a detrimental effect on the dynamic behaviour. The only known check valves of almost frictionless design are the Mannesmann Demag ring nozzle type check valves.

Dynamic Behaviour in Relation to Valve Size

Only a few experimental investigations have been carried out to determine the dependence of the dynamic behaviour on the valve size. In /2/ results of measurements are presented which were performed on check valves of 100 and 200 mm nominal size. The maximum reverse flow velocity can be found to be in about direct proportion to the valve size.
Shown in Fig. 8 are calculated characteristic dynamics for the Mannesmann Demag nozzle check valves of the type DRV-B. The spring forces are chosen in such a way that all valves have the same full opening velocity of 1.5 m/s. The results confirm the aforementioned dependency of the valve dynamics on the size.

If a system requires the installation of a quick-closing valve, a better performance can be obtained by using a valve of a smaller nominal size than that of the piping. The flow velocity and the deceleration will be increased, but the pressure rise due to valve closure will be smaller. This effect can be increased for spring-loaded valves because, on account of the higher flow velocity, they can be fitted with stronger springs.
However, on the other hand, when using smaller valves, the pressure loss will be increased, and therefore this method may be useful only for valves with a low pressure loss.

Check Valve in Line to Prevent Water Hammer

After a pump failure the pressure in a pipeline can drop to the vapour pressure of the liquid and vapour cavities may form. Large pressure rises can follow the cavity collapse. The most common surge protection devices such as surge vessels, surge shafts and air admission valves work in such a way that the pressure is kept above vapour pressure.

A not so common but useful aid can be to install a check valve in the line to prevent the cavity collapse by the return flow.

Fig. 9 shows corresponding calculated pressure-time graphs. The calculations were carried out on a single pipe system of a total length of 1.5 km, a water level elevation of 60 m and an initial flow velocity of 2.5 m/s. The location of the check valve was chosen downstream of the point where the greatest cavitation takes place (in this case about 400 m, downstream of the pump.). The graphs show that such a line valve is able to prevent waterhammer. It is neccessary for this purpose for the check valve to have a certain amount of leakage. Otherwise the pipe will stay partly filled and a pressure surge will occur after the restart of the pump. The one graph in Fig. 9 shows the result of calculations with a loss coefficient for reverse flow of a magnitude of 10^4. The lower the valve leakage the longer it takes for the pipe to fill up and a restart of the pump to become possible. On the other hand, the higher the leakage the less is the reduction in the pressure surge. For those reasons an application of this type of surge protection needs to be analysed carefully. It is favourable to install valves equipped with a small by-pass pipe with a throttle valve. There are valve manufacturers who have such valves in their standard range of products.

Conclusions

When choosing a certain type of check valve for a pipe system in which a high deceleration may occur, in the interest of avoiding check valve slam, a quick-closing type should be selected. Design parameters such as inertia and travel distance (displacement volume) of the moving parts can be helpful in estimating the dynamic behaviour of a certain valve.

Additionally, the manufacturers should be asked for information about the eigenfrequency and the so-called natural closing time. These are criteria giving a much better idea of the dynamic properties.
The manufacturers should also be able to give information about the pressure rise caused by their valves for a given value of flow deceleration.

The pressure loss in the valve may be small compared with the overall losses in the system, but nevertheless it should be kept in mind. Most valve types can be tuned by reducing the extent of the opening or by other design modifications to give a better dynamic performance, but normally this is connected with an increase in the frictional head loss. Further dynamic tests should be performed simultaneously with pressure loss measurements.

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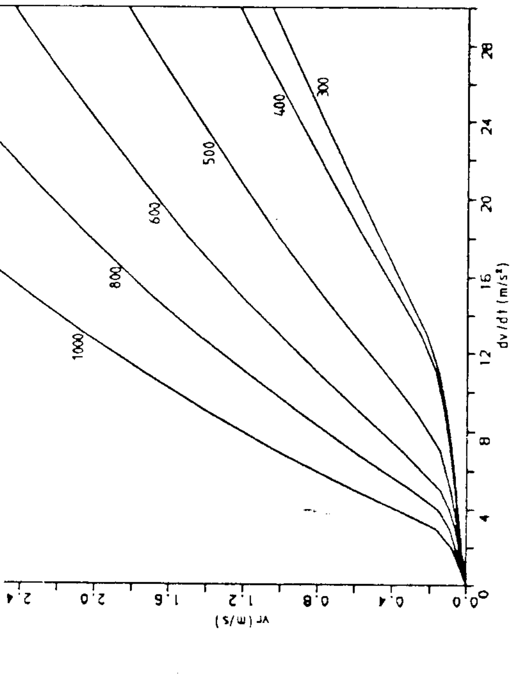


Fig. 8 Calculated dynamic characteristics of Mannesmann Demag check valves type DRV-B (parameter – nominal size in mm)

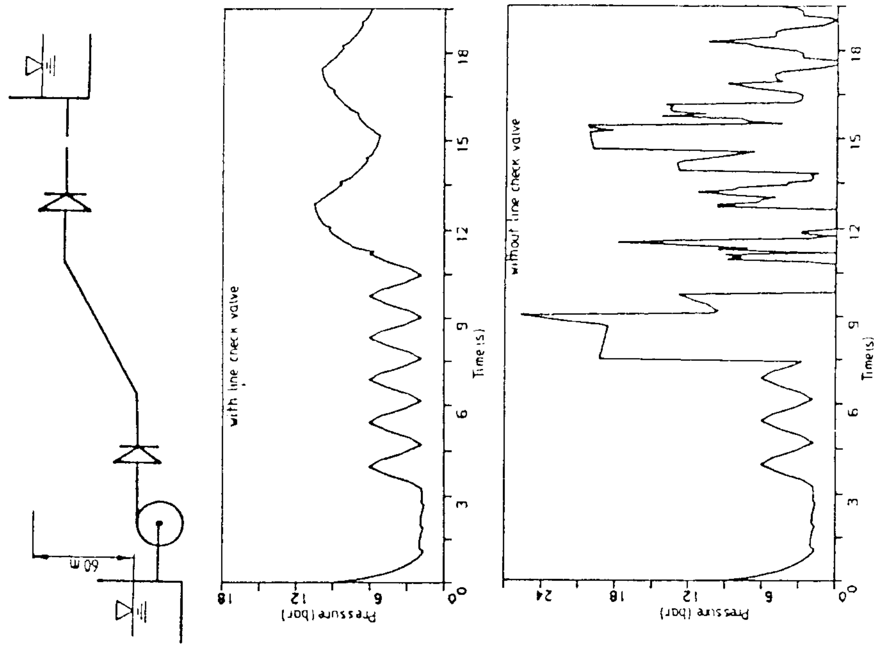


Fig. 9 Calculation of water hammer after pump

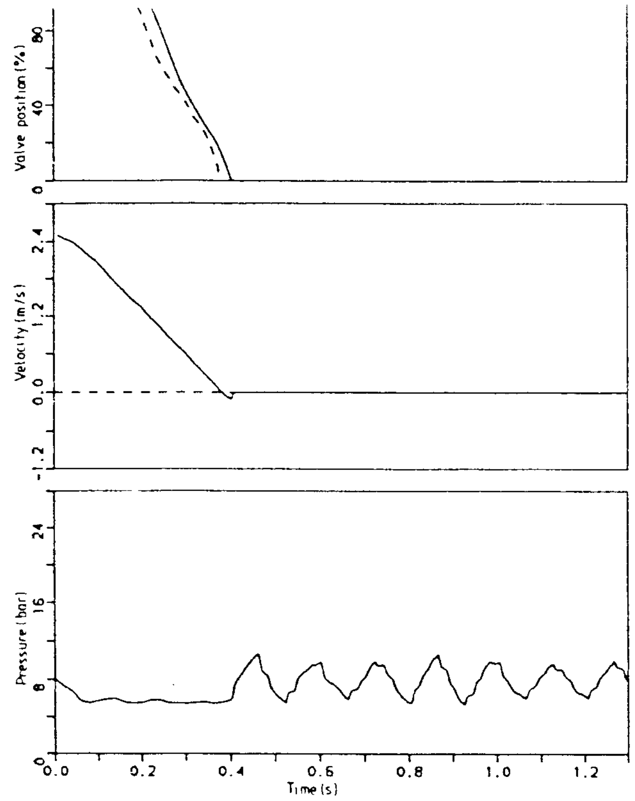


Fig. 4 Pressure transient calculation on a multi-pump system - ring type nozzle check valve

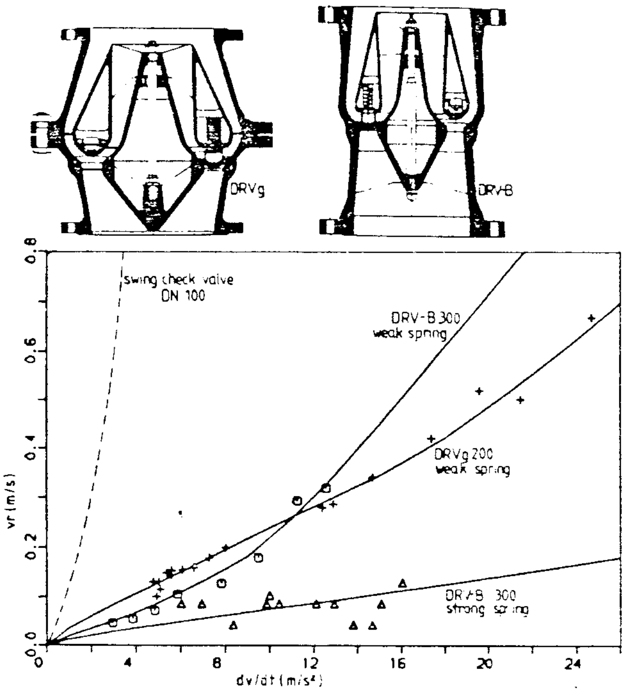


Fig. 6 Measured and calculated dynamic characteristic of Mannesmann Demag check valves

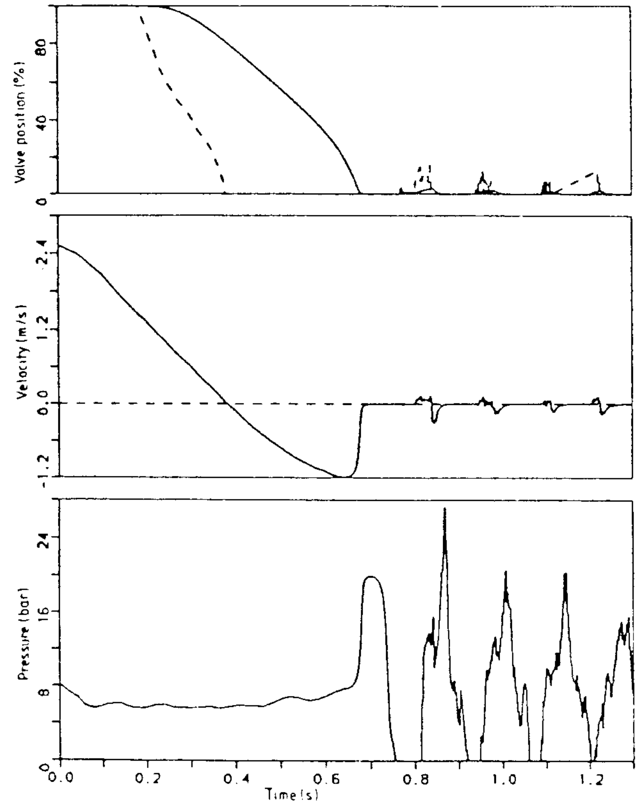


Fig. 5 Pressure transient calculation on a multi-pump system - swing type check valve

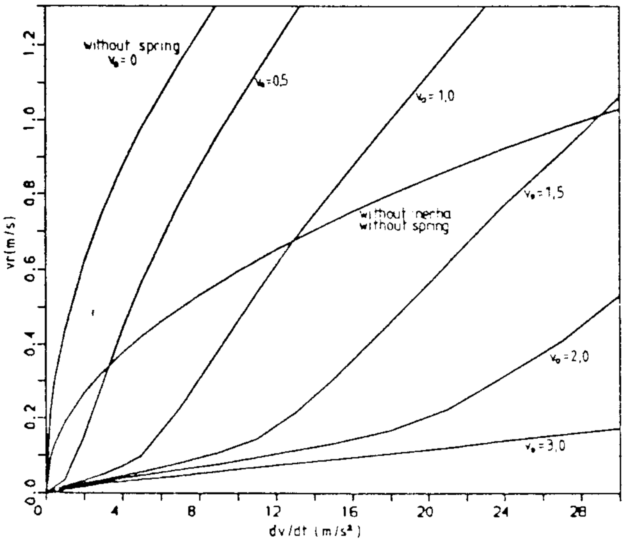


Fig. 7 Calculated dynamic characteristics of Mannesmann Demag check valve DRV-B 300 (parameter - full opening velocity in m/s)

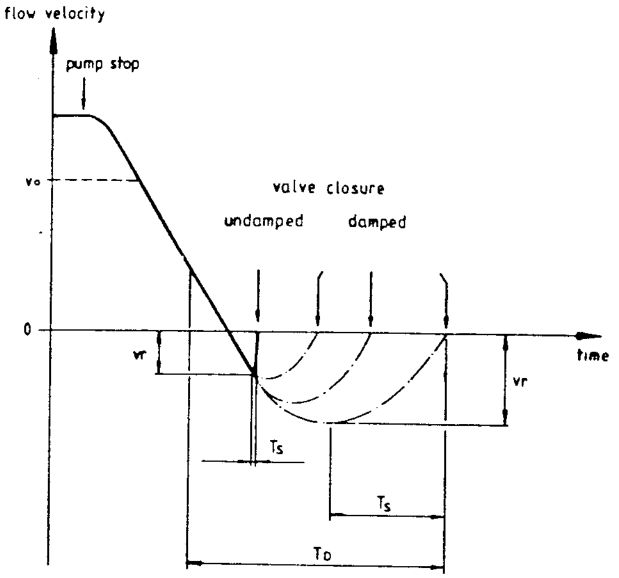
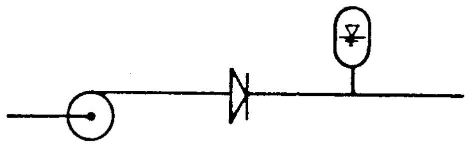
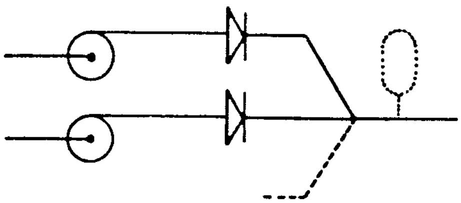


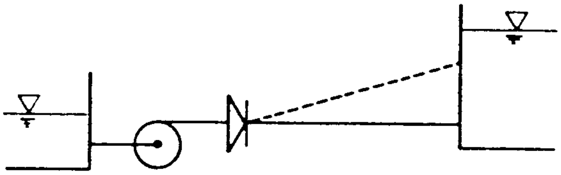
Fig. 1 Schematic flow-time history for check valves in decelerating flow



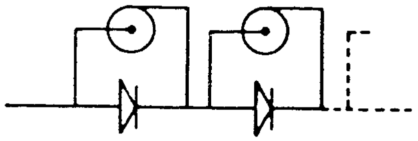
a)single pipe system with surge vessel or shaft



b)multi-pump system (with surge vessel or shaft)

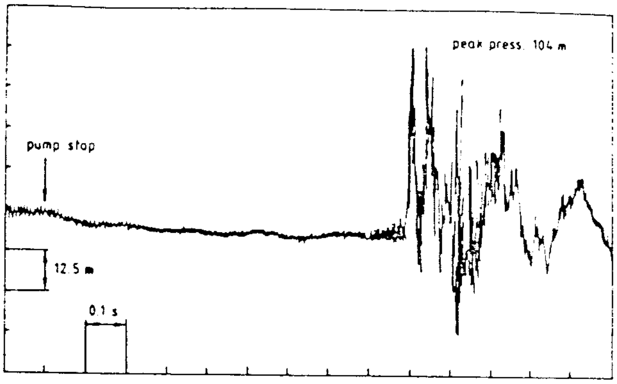


c)system with high elevation (vessel or shaft)

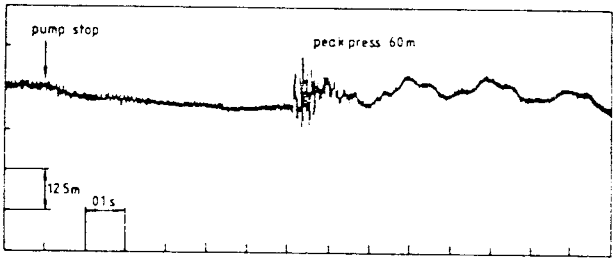


d)serial pump station

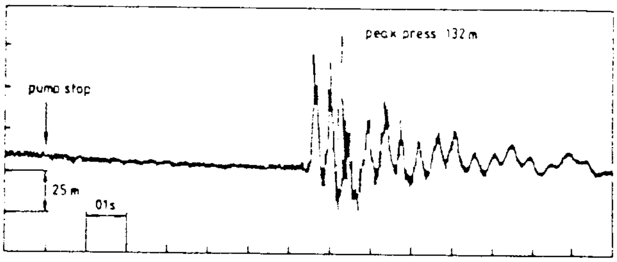
Fig. 2 Typical pipe systems with high flow deceleration at check valves



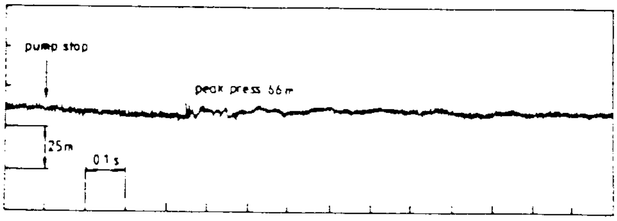
a)one pump running - swing check valve



b)one pump running - ring nozzle check valve



d)two pumps running - ring nozzle check valve



c)two pumps running - swing check valve

Fig. 3 Pressure-time histories immediately downstream of the check valve after pump failure

Proceedings of the 1st Midwestern Conference
on Fluid Dynamics, March 12-13, 1950
University of Illinois, publ. JW Edwards
Ann Arbor, 1951, p. 362-372.

Number 5

APPROXIMATE THEORY OF COMPRESSIBLE FLOW THROUGH REED VALVES FOR PULSE JETS

Paul Torda
University of Illinois

Abstract

A simple analysis of the flow process through automatically working reed valves is presented. The method used and the results obtained are compared with earlier work, the *exact* analysis. It is shown that the *approximate* theory is well suited to engineering applications.

Introduction

Reed valves are used for air intake on conventional pulse-jet engines. These valves are made of thin steel strips; they are clamped along one edge and are free to bend under the action of the pressure difference upstream and downstream of the valves. Thus, the movement of the valves is determined by the forward velocity of the unit, by the aero-thermodynamic process in the combustion chamber and the tail pipe, and by the elastic properties and the mass distribution of the reed valves. Fig. 1a is a schematic drawing of the pulse-jet engine; it also shows an enlarged view of a pair of clamped reed valves.

The exact analysis of the air inflow process has been presented in previous papers.[1,2,3] This exact analysis involves laborious computations and, therefore, is not suited for the first assessment of a new design. For such a purpose the present shorter, even though less exact, method is well suited.

There is one more reason why the *approximate* analysis is better suited for engineering purposes at present. In the *exact* method an indirect approach was employed in order to integrate the nonlinear partial differential equations in a closed form. This indirect approach determines the pressure distribution on the combustion-chamber side of the reeds from prescribed boundary values (initial, boundary, and transition conditions). A large number of numerical examples, covering all probable types of boundary values, would have to be solved in order to present the designer with combustion-chamber pressure distributions covering the domain of possible variations. The direct approach of the *approximate* theory enables the solution of each specific problem as it arises.

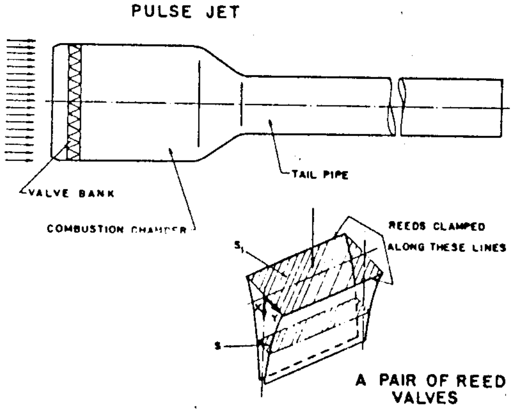


Fig. 1a

In the *approximate* analysis the bent shapes of the reeds during movement are determined first. This is done by prescribing the forcing function in the differential equation of the forced reed motion and by integrating this equation. Then, the inflow is treated as a quasi-steady nozzle flow between the reeds (i.e., the bent shapes of the reeds at specific time intervals during motion are taken as the shapes of nozzles) and the velocity, pressure, and density distribution of the flow are determined. For brevity, only the opening of the valves is treated, although the method is valid for the closing motion of the reeds as well.

The results of the approximate and the exact theories are compared. It should be noted, however, that the comparison is qualitative only, since in the exact theory supersonic velocities appeared, but in the approximate analysis only subsonic velocities were admitted; also, tapered reeds were used in the exact theory, but in the approximate theory constant mass distribution was assumed for the sake of simplicity.

Analysis

In the analysis the theory of non-viscous, compressible flow is used. It is assumed that the flow upstream of the valves is parallel and that the change of state of the gas is isentropic. As stated in the introduction, a quasi-steady flow between the reeds is considered at specific time intervals. The reeds are treated as cantilever beams of known mass distribution and elastic properties. Here, for simplicity, constant mass distribution is assumed. In order to reduce the shedding of vortices and thus reduce the losses at the free end of the reed, it is necessary that the pressure at both sides of the reeds be equal at this point at any time. Thus, the resultant pressure distribution along the reeds can be assumed to be triangular (Fig. 1b).

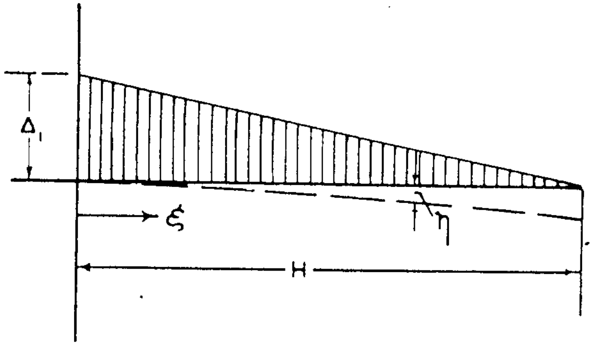


Fig. 1b. Reed Loading

Any other distribution of the resultant pressure forces can be ascertained without modifying the method of analysis.

The pressure difference along the reeds $[\Delta(x,t)]$ is the forcing function of the reed motion. Once this function is determined, the equation of the reed motion can be integrated by standard methods.[5] The solution of this equation gives the reed shapes at any time. Then, the use of the continuity equation, of the Bernoulli equation, and of the equation of change of state of the gas allows the determination of the flow variables.[6] The equation of forced motion of cantilever reeds is

$$\frac{\partial}{\partial \xi^2} EI \frac{\partial^2 \eta}{\partial \xi^2} + m \frac{\partial^2 \eta}{\partial t^2} = f(\xi,t), \tag{1}$$

where EI is the bending stiffness of the reeds
- m is the mass of the reed per unit of length
- ξ and η are the space variables along and perpendicular to the reeds
- t is the time variable
- $f(\xi,t)$ is the forcing function

In general, the bending stiffness and the mass distribution are known functions of ξ. However, for simplicity's sake, they were assumed constant in the present analysis. Therefore, Eq 1 can be written as

$$EI \frac{\partial^4 \eta}{\partial \xi^4} + m \frac{\partial^2 \eta}{\partial t^2} = f(\xi,t). \tag{2}$$

If the time-variable load per unit of length, the forcing function, is given by

$$f(\xi,t) = \Delta(\xi) e^{\omega t}, \tag{3}$$

the Eq 2 admits a solution of the form

$$\eta(\xi,t) = g(\xi) e^{\omega t}, \tag{4}$$

and $g(\xi)$ must satisfy the equation

$$EI \frac{d^4 \eta}{d \xi^2} + m\omega^2 \eta = \Delta(\xi), \tag{5}$$

By using the notation

$$\beta^4 = \frac{m\omega^2}{EI}, \tag{6}$$

Eq 5 becomes

$$\frac{d^4 \eta}{d\xi^4} + \beta^4 \eta = \frac{\Delta(\xi)}{EI}. \tag{7}$$

The solution of the homogeneous equation corresponding to Eq 7 is

$$h(\xi) = e^{\frac{\beta\xi}{\sqrt{2}}} \left[A \cos\left(\frac{\beta\xi}{\sqrt{2}}\right) + B \sin\left(\frac{\beta\xi}{\sqrt{2}}\right) \right] + e^{-\frac{\beta\xi}{\sqrt{2}}} \left[C \cos\left(\frac{\beta\xi}{\sqrt{2}}\right) + D \sin\left(\frac{\beta\xi}{\sqrt{2}}\right) \right]. \tag{8}$$

If $\Delta(\xi)$ is given (see Fig. 1b) by

$$\Delta(\xi) = \Delta_1 \left(1 - \frac{\xi}{H}\right), \tag{9}$$

a particular solution of Eq 7 is

$$k(\xi) = \frac{\Delta_1}{m\omega^2} \left(1 - \frac{\xi}{H}\right), \tag{10}$$

and the general solution of equation is

$$g(\xi) = h(\xi) + k(\xi). \tag{11}$$

The boundary conditions to be satisfied by $g(\xi)$ are

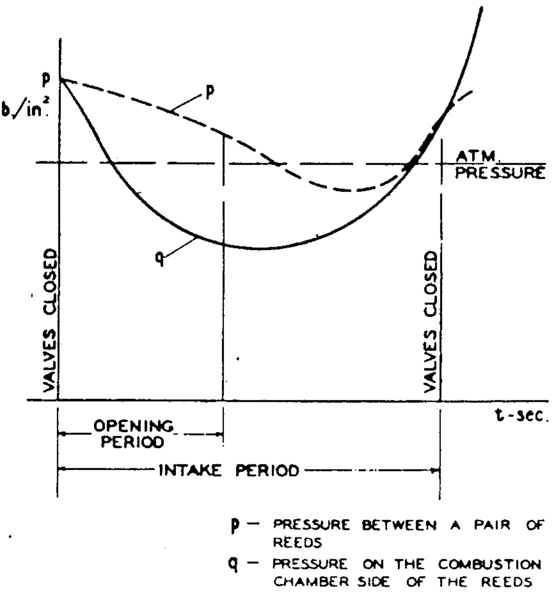


Fig. 2. Pressure Variation on Both Sides of the Reeds at $x = 0$

$$\left.\begin{array}{lllll} \text{for} & \xi = 0 & g(\xi) = 0 & \text{and} & \dfrac{dg}{d\xi} = 0 \\ \text{for} & \xi = H & \dfrac{d^2 g}{d\xi^2} = 0 & \text{and} & \dfrac{d^3 g}{d\xi^3} = 0 \end{array}\right\} \qquad (12)$$

From conditions (12) the four arbitrary constants of integration can be determined as

$$\left.\begin{array}{l} A = -(C + \lambda_2) \\ B = 2C - D + \lambda_2 \\ C = -\dfrac{1}{Q}[2e^{-2\alpha}\cos\alpha(\lambda_1 \sin\alpha + \lambda_2 \cos\alpha) + \lambda_1(1 + e^{-2\alpha})] \\ D = -\dfrac{1}{Q}[2\lambda_1 - \lambda_2(1 + e^{-2\alpha}) + 2e^{-2\alpha}\sin\alpha(\lambda_1 \sin\alpha + \lambda_2\cos\alpha)] \end{array}\right\} (13)$$

where

$$\lambda_1 = \frac{\Delta_1}{m\omega^2} \qquad \alpha = \frac{\beta H}{\sqrt{2}}$$

$$\lambda_2 = \lambda_1\left[1 + \frac{1}{\alpha}\right] \qquad Q = 1 + 2e^{-2\alpha} + e^{-4\alpha} + 4e^{-2\alpha}\cos^2\alpha .$$

The equations describing the flow in a nozzle, and therefore between a pair of reeds at any time (t), are:

The continuity equation is

$$\rho u s = \rho_n u_n s_n \qquad (14)$$

The Bernoulli equation is

$$u^2 + 2\phi = u_n^2 + 2\phi_n \qquad (15)$$

The equation of isentropic change of state is

$$\frac{p}{p_n} = \left(\frac{\rho}{\rho_n}\right)^{\gamma}, \qquad (16)$$

where u is the flow velocity
ρ is the fluid density
p is the fluid pressure
s is the cross-sectional area between a pair of reeds
γ is the adiabatic constant

$$\phi = \int \frac{dp}{\rho} = \frac{\gamma}{\gamma - 1}\frac{p_o}{\rho_o}\left(\frac{p}{p_o}\right)^{\frac{\gamma-1}{\gamma}} \quad \text{is the enthalpy}$$

subscript zero denotes stagnation conditions
subscript η denotes any specified station along x

Let subscript 1 denote the entrance section, $x = 0$, and let

$$U = \frac{u}{u_1} \qquad P = \frac{p}{p_1} \qquad R = \frac{\rho}{\rho_1} \qquad S = \frac{s}{s_1} \qquad (17)$$

be non-dimensional velocity, pressure, density, and area variables, then the inflow velocity, U, and the inflow pressure, P, can be calculated from the following two equations:

$$\frac{1}{S^2} = \left[\frac{2\gamma}{u_1^2(\gamma - 1)}\frac{p_o}{\rho_o}^{\frac{\gamma-1}{\gamma}}\left(1 - P^{\frac{\gamma-1}{\gamma}}\right) + 1\right] P^{\frac{2}{\gamma}} \qquad (18)$$

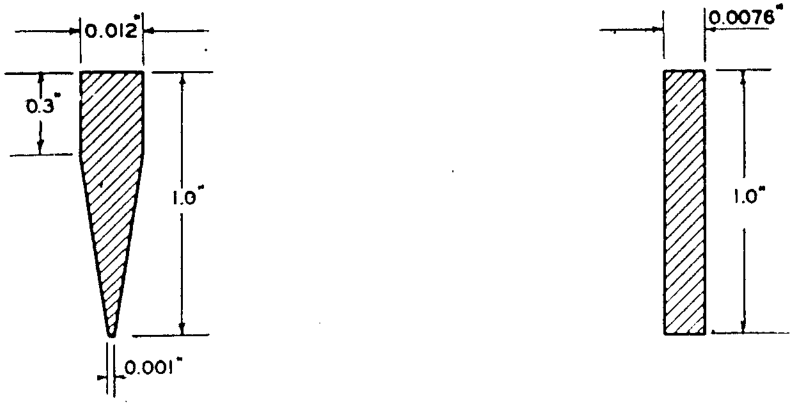
$$U^2 = \frac{2\gamma}{u_1^2(\gamma - 1)}\frac{p_o}{\rho_o}\left(\frac{p_1}{p_o}\right)^{\frac{\gamma-1}{\gamma}}\left(1 - P^{\frac{\gamma-1}{\gamma}}\right) + 1 \qquad (19)$$

and the relation between u_1 and p_1 is given by

$$u_1^2 = \frac{2\gamma}{\gamma - 1}\frac{p_o}{\rho_o}\left[1 - \left(\frac{p_1}{p_o}\right)^{\frac{\gamma-1}{\gamma}}\right] \qquad (20)$$

Eqs 18 and 19 admit solutions for both subsonic and supersonic velocities. Here the velocities were restricted to the subsonic region, since it is believed that because of frictional and other losses supersonic velocities will not occur in actual cases.

At a selected station, say at the entrance, x = 0, the pressure variation with time [q = q(t)] in the combustion chamber can be given from measured data or from analysis of the aero-thermodynamic process in the combustion chamber. Then the intake pressure has to be determined at the same station as a function of time [p = p(t)]. The



EXACT THEORY

APPROXIMATE THEORY

Fig. 3. Reed Dimensions Used in Numerical Examples

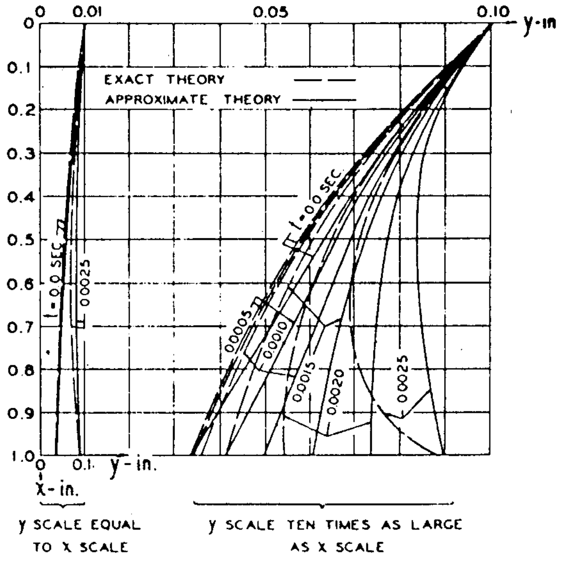


Fig. 4. Reed Deflections

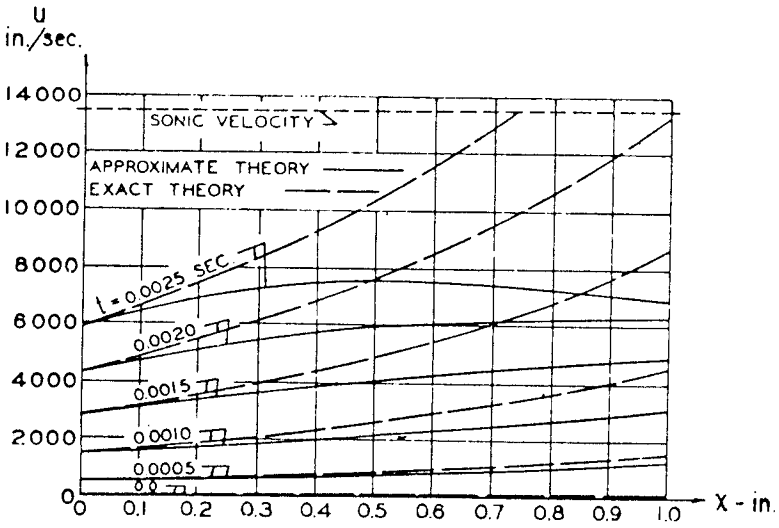


Fig. 5. Inflow Velocity Distribution between a Pair of Reeds

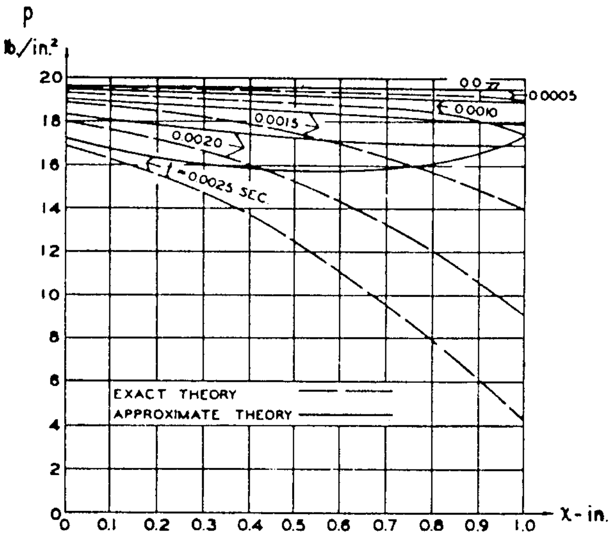


Fig. 6. Pressure Distribution between a Pair of Reeds

intake pressure (p) should be so chosen that the resultant pressure acting on the valves ($\Lambda = p - 1$) should be large during opening and small during closing of the valves. This choice will increase the amount of air intake in each cycle and will reduce the impact force during seating of the valves as well. Thus, the appropriate choice of $p_1 = p_1(t)$ makes it possible to determine the flow variables. Characteristic variation of p and q is shown in Fig. 2.

The transformation from the ξ, η coordinates to the x, y coordinates can be made analytically or graphically.

Numerical Example

The numerical example included has been selected to conform in general to one of the examples of the *exact* analysis.[1,2] The deviations of the results are due to the different mass distribution of the reeds (see Fig. 3).and to the fact that no supersonic inflow velocities were admitted in the *approximate* analysis, while in the *exact* analysis such velocities appeared. Instead of the flow pressures, the the flow velocities at the entrance of the valves were matched. Figs. 4, 5, and 6 show the reed deflections, the inflow velocity variation, and the inflow pressure distribution, respectively, for both the *approximate* and the *exact* theories. Fig. 7 gives the pressure variation on the combustion-chamber side of the reeds for the *approximate* theory only.

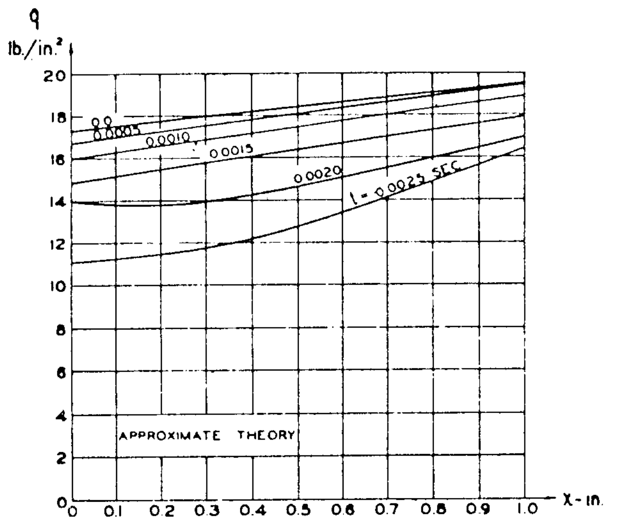


Fig. 7. Pressure Distribution on the Combustion-Chamber Side of the Reeds

Concluding Remarks

On concluding the analysis it should be pointed out that the *approximate* analysis seems to yield results sufficiently accurate for design purposes. However, accurate experiments, lacking at present, should be carried out to determine the flow phenomena upstream and downstream of the valves during inflow.

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Acknowledgements

The author wishes to thank the Office of Naval Research for the release of the material presented. Miss Ann Gunsolns and Mr. J. H. Brick carried out the numerical work very conscientiously. The analysis was made at Polytechnic Institute of Brooklyn in 1948.

10-217-0762

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DATE OF ORDER 10-18-88
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FILING/ISSUE DATE 02-11-36

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053 **1934** **DATED** FEB 11 1938 ~~55/13~~

DIV. 189 **(EX'R'S BOOK)** J-106-15

Name BOB NEAL

ARKADELPHIA,

~~***State of***~~ ARK~~ANSAS~~

Invention ~~ENGINES~~ Compressor Unit

ORIGINAL	**RENEWED**
APPLICAT[illegible]D COMPLETE JAN 9, *1934*	16
Peti[illegible]cification, Oat[illegible] Fee $30, [illegible] Drawings, JAN 9, *1934*	

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[illegible] **1934** **DATED** FEB 11 1936

DIV. ~~4~~ ~~18~~ 9

(EX'R'S BOOK) ~~55~~ ~~13~~

J-106-15

Name BOB NEAL,

ARKADELPHIA,

~~*State of*~~ ARK~~ANSAS~~

Invention

~~ENGINES~~ Compressor Unit

ORIGINAL		RENEWED
APPLICATION FILED COMPLETE	JAN 9, *1934*	16
Petition, Specification, Oath, First Fee $30, 3 *sheets Drawings,*	JAN 9, *1934*	

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DIV. ~~4~~ ~~18~~ 9

(EX'R'S BOOK) J-106-15 ~~18~~

Name BOB NEAL

ARKADELPHIA,

ARK~~ANSAS~~

Invention ~~ENGINES~~ Compressor Unit

ORIGINAL

FILED COMPLETE JAN 9, 1934

Specification, First Fee $30, Sheets Drawings, JAN 9, 1934

and passed for Issue Sec. 17, 1935

Abrams Exr. Div. 9

of Allowance Dec 17, 1935 By Commissioner.

Final Fee $30 Jan 16, 1936

RENEWED

16

Reexam'd and passed for Issue

Exr. Div.

Notice of Allowance By Commissioner

Final Fee

Attorney FRANK O. PARKER 202 BARRISTER BLDG CITY.

Associate Attorney

No. of Claims Allowed 1 Print Claim 1 in O. G. Class 230-18

Title as Allowed Compressor Unit

APPLICATION FOR U. S. LETTERS PATENT
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MAIL DIVISION
JAN 9 34

PETITION

705964

JAN K 1024 30.00

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To the Commissioner of Patents:

Your petitioner (1) Bob Neal, citizen of the United States of America, residing at Arkadelphia, in the county of Clark, and State of Arkansas, and whose post-office address Arkadelphia, Arkansas, prays that Letters Patent may be granted to him for the improvement in Compressor Unit ~~ENGINES,~~ set forth in the annexed Specification.

And he hereby appoints (2) Frank O. Parker, 202 Barrister Building, of Washington, D.C., Register No. 6150 Attorney with full power of substitution and revocation to prosecute this application, to make alterations and amendments therein, to sign his name to the drawings, to receive the Letters Patent, and to transact all business in the United States Patent Office connected therewith.

Signed at ARKADELPHIA, in the County of CLARK and State of ARKANSAS this 12th day of December 1933

(Sign here First name in Full.) Bob Neal

(1) In case of assignment of the invention, application and oath must be made by actual inventor, even if patent issue to assignee.
(2) If the power of attorney be given to a firm, the full name of each member of firm must be given, or it will be ignored.

SPECIFICATION.

To All Whom it May Concern:

BE IT KNOWN, That Bob Neal, citizen of the United States of America, residing at Arkadelphia, in the County of Clark, and State of Arkansas, has invented certain new and useful improvements in Compressor Unit

10243

The invention relates to ~~an engine~~ a compressor construction, and more particularly to a combination fluid operated engine and compressor.

The primary object of the invention is the provision of ~~an engine~~ a compressor of this character, wherein there is arranged an automatically counter balanced crank shaft and fluid equalizers within a storage tank, which makes it possible for the said engine to operate on constant reserve tank pressure so as to actuate additional equipment, the pistons for the engine being also automatically balanced and suspended when the said engine is in operation.

Another object of the invention is the provision of an engine of this character, wherein the same is operated from air under pressure, the said air being supplied by compressors, these being in bank with the engine construction.

A further object of the invention is the provision of an engine of this character, wherein the same is of novel construction, as the engine proper and the compressors are operated from the same crank shaft which is of the automatically balanced type, so that high efficiency is attained.

A still further object of the invention is the provision of an engine of this character, which is comparatively simple in construction, thoroughly reliable and efficient in its operation, strong, durable, and inexpensive to manufacture.

With these and other objects in view the invention consists in the features of construction, combination and

10244

arrangement of parts as will be hereinafter more fully described, illustrated in the accompanying drawings, which disclose the preferred embodiment of the invention, and pointed out in the claim~~s~~ hereunto appended.

In the accompanying drawings:

Figure 1 is a perspective view of the engine constructed in accordance with the invention.

Figure 2 is a vertical transverse sectional view through the compressor part of the engine.

Figure 3 is a vertical sectional view through the power part of the engine.

Figure 4 is a detail elevation of the crank~~ed~~ shaft of the engine.

Figure 5 is an enlarged sectional view through one of the electric heaters for the engine.

Figure 6 is a vertical longitudinal sectional view through the air storage tank including the equalizer.

10245

Similar reference cheracters indicate corresponding parts throughout the several views in the drawings.

Referring to the drawings in detail the engine in its entirety comprises a cylinder block 10 having formed therein the series of compressor cylinders 11 and the ~~power~~ both cylinders 12, respectively, the block 10 being of the V-type and closing the upper ends of said cylinders are the removable heads 13 and 14, respectively, which are secured in place by head bolts 15, as is conventional. Beneath the block 10 is the crank case 16, which at opposite sides carries the detachable plates 17, these being held in place by fasteners 18 and such plates are seated so as to be leak proof. The block 10 is chambered to provide a water jacket 19 about the cylinders, while at the forward end of the said block ~~is a~~ are water pumps 20 circulating water through the inlet pipe 21 which leads into the jacket and letting said water out therefrom through the outlet pipe 22 leading from said water jacket. Next to the pumps 20 is a fan 23 operated from a belt 24 ~~which also drives~~ the pumps.

14

Working within the cylinders 11 are the reciprocating pistons 25, their ~~stems~~ rods 26 being slidable through packing glands 27 and ~~are~~ fixed to ~~runners~~ crossheads 28, which are slidably mounted upon guides 29 secured within the crank case 16 to opposite side walls thereof. These ~~runners~~ crossheads 28 are fitted with wrist pins 30 pivotally connecting therewith the connecting rods 31 which by the bearings 32 are engaged with their cranks 33 of a ~~an automatically~~ counter balanced crank shaft 34, which is mounted in supports

35 arranged in the crank case 16, the shaft being supplied with the required bearings 36.

The inner ends of the cylinders 11 are fitted with inner end heads 37, which are provided with air intake ports 38, these being fitted with spring ball inlet checks 39, the air having admission through passages 40 opening exteriorly of the block 10. The glands 27 are associated with the heads 37.

The heads 13 and 37 are provided with the compressed air outlets 41 and 42, respectively, these being fitted with spring ball checks 43, the heads 13 being also provided with the central air inlets 44, which are fitted with spring ~~ball~~ checks 45. By couplings 46 are attached to the air outlets 41 and 42 ~~are~~ the outlet feed pipes 47 and 48, respectively, these leading to a main conduit 49 located in the center channel 50 in said block 10.

At the rear end of the block 10 and on the shaft 34 is the fly wheel 51, this being of conventional type.

Working within the cylinders 12 are pistons 52, their ~~stems~~ rods 53 sliding through packing glands 54 and ~~are~~ fixed in ~~runners~~ crossheads 55 slidably mounted upon guides 56 which are secured within the crank case 16 at opposite side walls thereof. The ~~runners~~ crossheads 55 carry wrist pins 57 connecting therewith connecting rods 58, these being engaged by bearings 59 with their respective cranks 60 of the crank shaft 34, the inner ends of the cylinders 12 being also closed by inner heads 61 with which are associated the glands 54.

On the cylinders 12 are slide valve chests 62 in which are the slide valves 63, these being operated by throw rods 64 actuated by cams 65 and such valves controlling the air admission and exhaust of air to and from the cylinders 12 through the ports

wa 66 and 67, ~~respectively,~~ and these valves 63 are provided with the ports 68 for the delivery of air under pressure from the inlet passages 69 common to a lead 70 from a compressed air storage tank 71.

The bottom of the crank case 16 is fitted with a removable plate 72 which is secured in place by fasteners 73, and when this plate is removed access can be had to the crank shaft 34 and the bearings for the engine, as well as other parts within said crank case, as should be obvious.

Leading into the cylinders 11 are the passages 74 of a lubricating system (not shown).

The storage tank 71 for the compressed air includes

wa therein ~~an equalizer~~ a double check discharge nozzle 75, this being supported by a member 76 and leading to this equalizer is an air inlet pipe 77 which has the communication 78 with the chamber 79 formed by said tank. In the equalizer 75 are the spaced spring ball checks 80 and 81, respectively, one being for the inlet side and the other for the exhaust or outlet side of said equalizer. This pipe 77 is connected with the main conduit 49, while a pipe 82 is connected with the leads 70, the tank being also fitted with an automatic relief valve 83 of any approved type.

About the pipes 70 for the passages 69 are the electric heating units 84 which are for the purpose of heating the air under pressure above a freezing temperature when delivered from the tank 71 to the cylinders 12.

Supported on the block 10 is an electric generator 85 which is driven from the shaft 34 through a belt 24 and this generator is included in an electric circuit which also has the heaters 84 so that these will operate from current furnished by said generator.

The storage tank 71 with the equalizer are so constructed that it is possible to pump air into the said tank with a tank pressure of two hundred pounds, while the compressors are only pumping against fifteen pounds or atmospheric pressure. Outside air pressure source can be coupled with the tank to augment that pressure derived from the cylinders 11 of the engine.

10243

What is claimed is:

Add C'pe 1

1. An engine of the character described, comprising a block having cylinders, pistons operating in said cylinders, certain of the cylinders and their pistons forming a fluid compressor unit and the other cylinders and pistons forming a power unit, means for storage of compressed fluid from the compressor unit, and means for delivering compressed fluid from said means to the power unit.

2. An engine of the character described, comprising a block having cylinders, pistons operating in said cylinders, certain of the cylinders and their pistons forming a fluid compressor unit and the other cylinders and their pistons forming a power unit, a storage tank for the compressed fluid from the compressor unit, means for delivery of compressed fluid from the tank to the power unit, and a fluid equalizer within the tank. 1-C2

per A

3. An engine of the character described, comprising a block having cylinders, pistons operating in said cylinders, certain of the cylinders and their pistons forming a fluid compressor unit and the other cylinders and their pistons forming a power unit, a storage tank, means for delivery of compressed fluid from the compressor unit to the tank, means for feeding compressed fluid from the tank to the power unit, a fluid equalizer within the tank and coacting with the first named means, and a heater connected with the last named means. 2+C2

4. An engine of the character described, comprising a block having cylinders, pistons operating in said cylinders, certain of the cylinders and their pistons forming a fluid

10250

compressor unit and the other cylinders and their pistons forming a power unit, a storage tank, means for delivery of compressed fluid from the compressor unit to the tank, means for feeding compressed fluid from the tank to the power unit, a fluid equalizer within the tank and coacting with the first named means, a heater connected with the last named means, a balanced crank shaft common to and connected with said pistons, and valve mechanisms for the inlet and outlet fluids to and from said units, one mechanism being controlled by said crank shaft.

10254

In Testimony Whereof I affix my signature.

Bob Neal

(Sign here, First name in full.)

OATH

STATE OF ARKANSAS
COUNTY OF CLARK } ss:

Bob Neal

the above-named petitioner, being duly sworn, deposes and says that he is a citizen of (1) the United States of America, and resident of Arkadelphia, Arkansas, that he verily believes himself to be the original, first, and (2) sole inventor of the improvements in ENGINES, described and claimed in the annexed specification; that he does not know and does not believe that the same was ever known or used before his invention or discovery thereof, or patented or described in any printed publication in any country before his invention or discovery thereof, or more than two years prior to this application, or in public use or on sale in the United States for more than two years prior to this application; that said invention has not been patented in any country foreign to the United States on an application filed by him or his legal representatives or assigns more than twelve months prior to this application; and that no application for patent on said improvement has been filed by him or his representatives or assigns in any country foreign to the United States, ~~except as follows~~: (3)

☞ (Sign here First name in Full.) Bob Neal

Sworn to and subscribed before me this 12th day of December 1932

(Official Signature.)

Notary Public
(Official Character.)

[Seal here, to be impressed in paper.]

My Commission expires Mar. 15, 1935.

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Print of drawing as originally filed.

Fig. 5. Eng

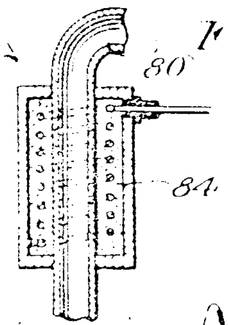
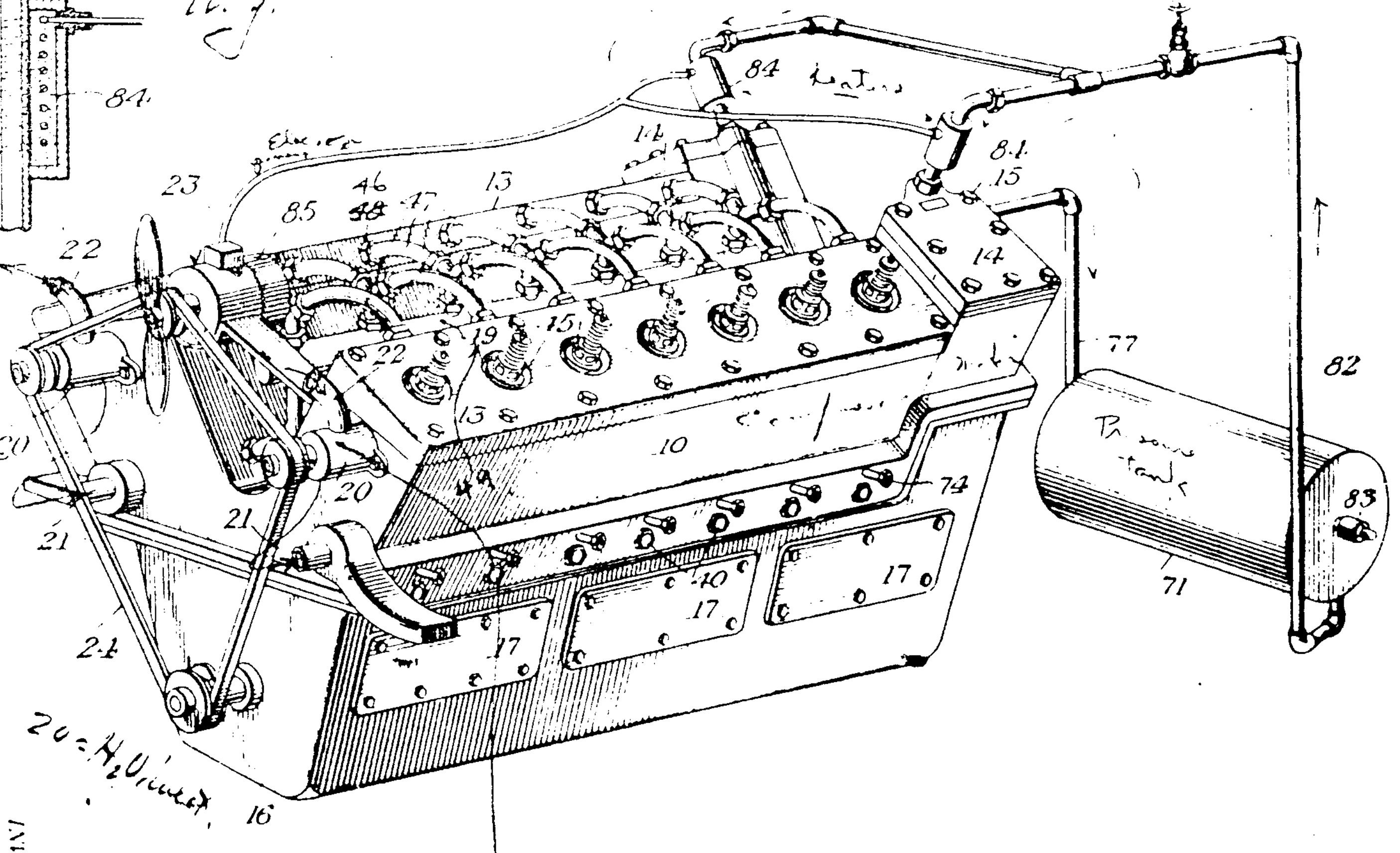


Fig. 1. — Eng



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INVENTOR.
Bob Neal
BY
ATTORNEY.

9

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Print of drawing as originally filed.

Fig. 2.—Compressor

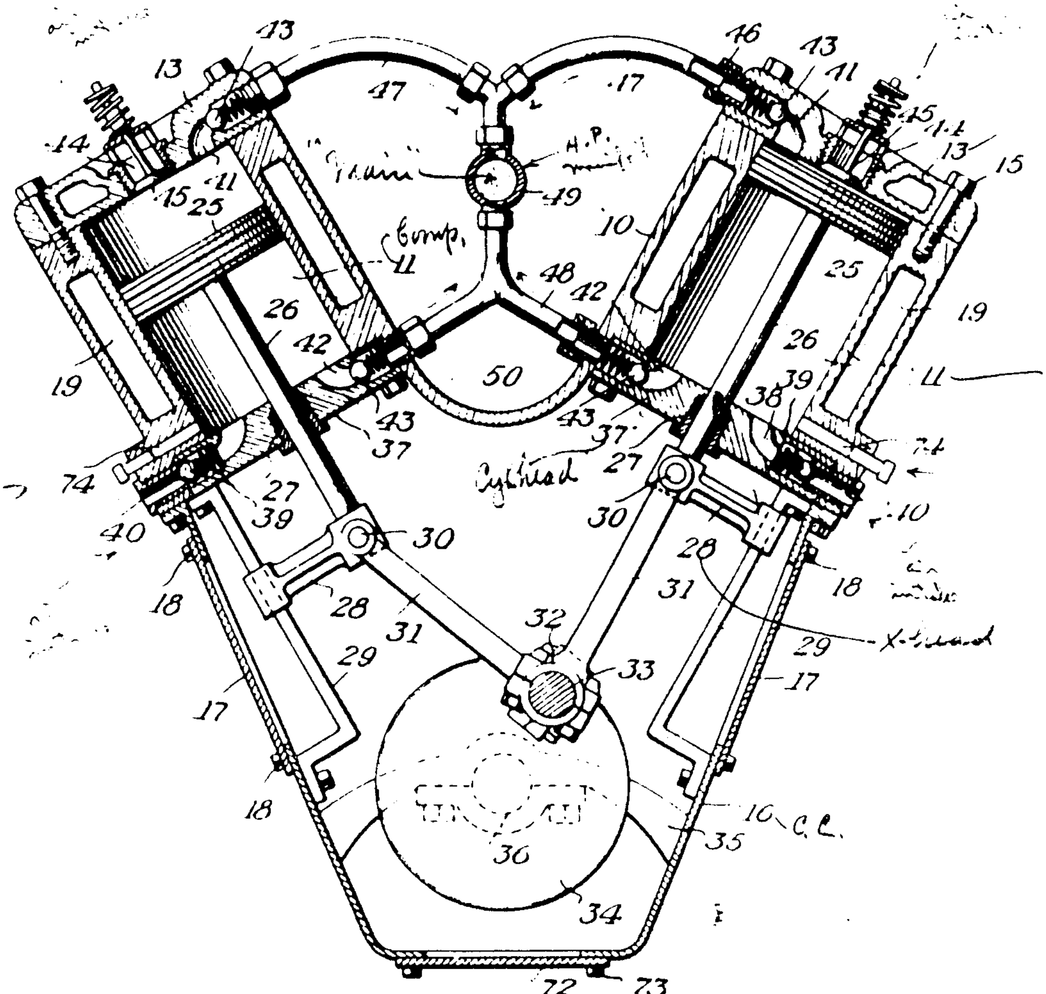

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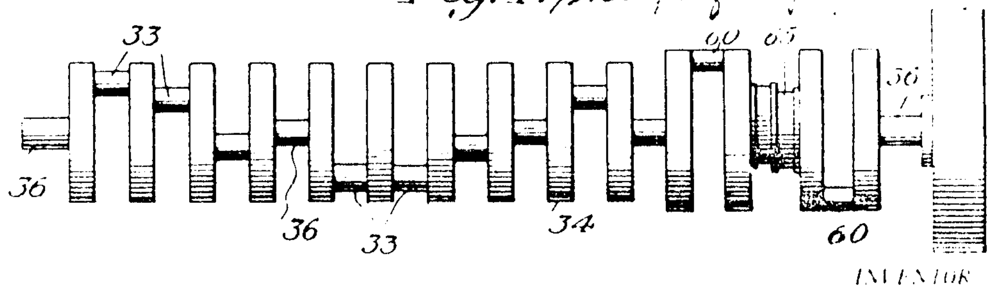
Fig. 4. "Steam engine"



INVENTOR

Bob Neal

BY

ATTORNEY

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Fig. 3.

(Engine)

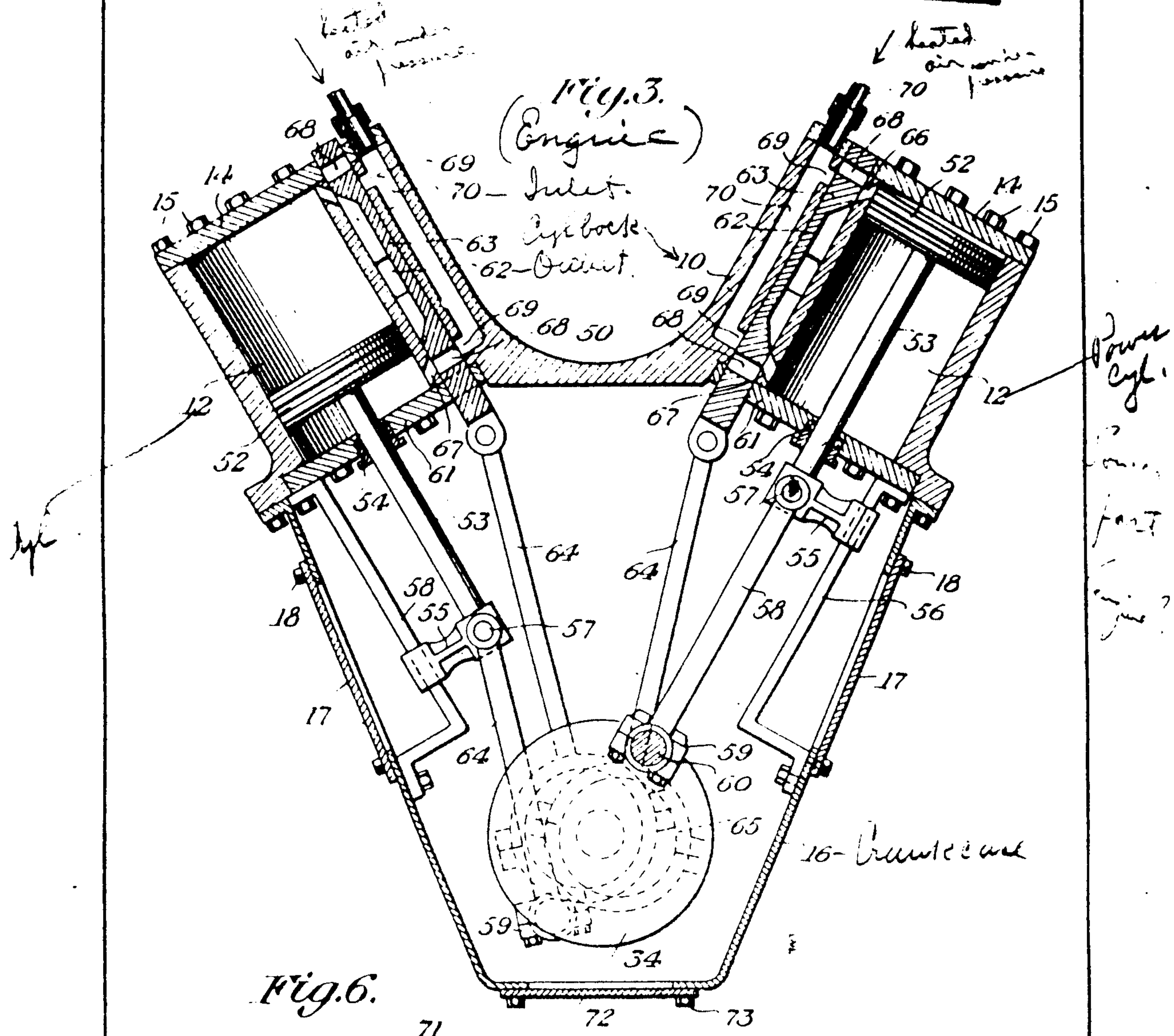
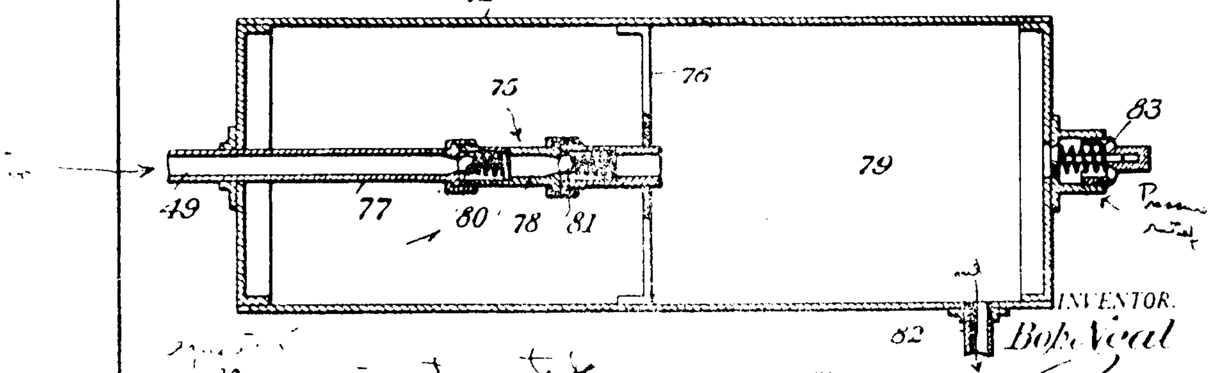


Fig. 6.



air storage tank & equalizer?

INVENTOR.
Bob Neal
BY
ATTORNEY

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DEPARTMENT OF COMMERCE
UNITED STATES PATENT OFFICE
WASHINGTON

All communications respecting this application should give the serial number, date of filing, and name of the applicant

Please find below a communication from the EXAMINER in charge of this application.

Conway P. Coe
Commissioner of Patents.

11—6838

Frank O. Parker,
202 Barrister Bldg.,
City.

Applicant: Bob Neal,

Ser. No. 705,964,
Filed Jan.9,1934,
For Engines.

MAILED
MAR 12 1934

This case has been examined.

References made of record:

Kneedler	1,297,303	Mar. 13,1919	60-59
Flood	662,189	Nov.20,1900	60-59
Jenkins	868,441	Oct.15,1907	60-59 (x60-33)

✓ In Fig. 1, numeral 48 has been changed to 46; and, in Fig. 5, numeral 80 to 70.

✓ In Fig. 6, numeral 75 has been supplied with a lead line.

Applicant's first name appears to be an abbreviation. The petition and preamble are required to be amended to give his full first name; or, if "Bob" is such full name, a statement over applicant's signature to that effect is required. See Order #3140.

R

✓ **Page 3,** line 12, "cranked" has been changed to -crank-.

✓ **Page 4,** line 14, two pumps 20 are shown in Fig. 1; **line 21, "are"** before "fixed" has been canceled; lines 20, 21 and 23, "stems" and "runners" should be -rods- and -crossheads-, respectively. Line 25, nothing "automatic" is disclosed in the balanced crank shaft.

✓ Page 5, line 7, add the sentence -The glands 27 are associated with the heads 37.-; lines 11-12, the check valves 45 are not of the "ball" type; line 13, change "are" to -the-; line 19, "are" has been canceled; lines 19, 20 and 22, "stems" and "runners" should be changed to -rods- and -crossheads-, respectively; line 23, the term -being- should preferably be inserted before "operated".

Page 6, line 2, cancel "respectively"; line 21, "lead" has been pluralized; line 23, numeral 70 has been inserted after "pipes".

The "equalizer" 75 appears to be lacking in utility and inoperative to perform the function ascribed thereto. Fuller explanation of its alleged operation is required.

Applicant's system is inoperative and lacking in utility for the purpose set forth. A round robin circuit with no input of energy is disclosed. The air under pressure from a tank is supposed to cause an air engine to operate compressors which latter are connected to return the exhaust air back to the tank under pressure. The engine is also to drive an electric generator which is to supply the energy to heating coils arranged on the engine intake pipes. Because of inevitable losses (friction, heat radiation, slip, etc.) applicant's device cannot maintain itself in operation under no load, and much less can it supply energy for any useful purpose.

Accordingly, claims 1, 2, 3 and 4 are rejected as being drawn to a device which is inoperative and lacking in utility for the purpose described.

Claims 1, 2, 3, and 4, insofar as they define structure having utility, are further rejected as fully met in each of the patents to Kneedler, Flood and Jenkins.

All the claims are rejected.

M E Porter
Examiner.

JAK.

#4

MAIL DIVISION
AUG 31 34
U.S. PATENT OFFICE

IN THE UNITED STATES PATENT OFFICE.

In re: application of
Bob Neal,
ENGINES,
Filed January 9, 1934,
Ser. No. 705,964.
Div. 18. Room 4612.

AMENDMENT.

PATENT OFFICE
SEP 1 - 1934
DIVISION 18

Hon. Commissioner of Patents,
Washington, D.C.

SIR: -

In response to the office action of March 12, 1934 in the above entitled application I hereby amend the same as follows:

In the specification, page 4, line 14 after "block" cancel "is a" and insert "are"

Same line, cancel "pump" and insert "pumps"

Line 17 18 and also line 19 change "pump" to "pumps"

Line 20 cancel "stems" and insert "rods"

Lines 21 and 23 change "runners" to "crossheads"

Line 25 cancel "an automatically" and insert "a"

Page 5, line 7 after the period (.) insert "The glands 27 are associated with the heads 37."

Line 11 cancel "ball"

Line 13 cancel "are" and insert "the"

Line 19 cancel "stems" and insert "rods"

Lines 20 and 22 cancel "runners" and insert "crossheads"

Line 28 before "operated" insert "being"

Page 6, line 2 cancel "respectively"

Line 14 cancel "an equalizer" and insert "a double check discharge nozzle"

✓ Cancel the claims and insert:-

8. An engine of the kind described, comprising a V-shaped cylinder block providing cylinders, inner 37 and outer 13 end heads closing opposite ends of the cylinders and having air intakes 40, a crank 16 case fitted with the inner end heads of the cylinders, a counterbalanced crank shaft Fig 4 journaled in said case, pistons 25 working in the cylinders and having rods 26 extended into said case, crosshead guides 29 fitted to the sides of the case, crossheads connecting the rods with the guides, connecting rods operated by the crank shaft and pivoted to the rods of the pistons, and a system of pipes 47.48,49 arranged at the crotch of the V-shaped block and having lateral communication with the cylinders at opposite sides of the pistons therein for delivering fluid under pressure from said cylinders.

REMARKS.

Applicant has corrected the specification as has been suggested and required by the Examiner. It is believed that all objections to this specification have been overcome and particularly that part questioned.

Now with respect to the operativeness of this engine the applicant holds that he can demonstrate such working and if the Examiner insists thereon a time will be set for so doing. It is possible that the Examiner has overlooked the fact that outside energy is supplied to this engine and this fact is brought out in the specification

The oath in this case certifies that the applicant's first name is as signed thereto and thus that the first name is Bob.

In lieu of the original claims applicant presents herein a newly drafted claim and this calls for structure not taught by the references and therefore such references cannot be held as anticipatory thereof. In this the Examiner's attention is directed to the construction of the block with the crank case attached thereto and the sides of this case are supplied with crosshead guides traversed by crossheads which are carried by the rods of the pistons working within the cylinders formed by the block. These crossheads have pivoted thereto the connecting rods operated by the crank shaft which is journaled in the case. Furthermore, attention is called to the system of piping arranged at the crotch of the V-shaped block and the manner of communication with the cylinders of said block. These structural features are not shown or suggested by the references.

It is thought that this application is now in condition for allowance and an action to this end is respectfully requested at an early date.

Respectfully submitted,

Paul O. Parker

Attorney of record.

Washington, D.C.

August 31, 1934.

Address only
"The Commissioner of Patents,
Washington, D. C.,"
and not any official by name

DEPARTMENT OF COMMERCE
UNITED STATES PATENT OFFICE
WASHINGTON

application should give the serial number,
date of filing, and name of
the applicant

Oct. 30, 1934.

[illegible]ease find below a communication from the EXAMINER in charge of this application.

Conway P. Coe
Commissioner of Patents.

11—8023

Frank O. Parker
202 Barrister Building
Washington, D.C.

Applicant: B. Neal

Ser. No. 705 964
Filed Jan. 9, 1934
For Engines

Responsive to amendment of Aug. 31, 1934.

References added:

Shimpf	1144 437	June 9, 1915	230-187 x-r
Adenot	1398 355	Nov. 29, 1921	230-187 x-r
Christensen	1369 618	Feb. 22, 1921	230-58 x-r
Reall	1230 219	June 19, 1917	230-56
Pettee et al.,	637 661	Nov. 21, 1899	230-56
Rayfield	1846 656	Feb. 23, 1932	230-187

It is now not clear whether applicant is claiming an engine or a motor compressor unit in which the patentable novelty, if any, resides in compressor structure as disclosed in Fig. 2.

In the former view, the claim now presented must again be rejected on the grounds and for the reasons fully set forth on the second page of the former Office letter, i.e., as for an inoperative system.

In the latter view, the claim now presented is rejected as failing to set forth the invention with the particularity and distinctness required by section 4888 R.S.

Accordingly, claim 5 is rejected.

References are made of record as further showing the state of the art.

R. E. Adams
Examiner.

Trans. to Div. 7,

APR 29 35

DIVISION 5 PAPER NO. 6
APR 30 1935
U. S. PATENT OFFICE

IN THE UNITED STATES PATENT OFFICE.

In re: application of
Rob Neal,
Engines,
Filed January 9, 1934,
Ser. No. 705,964.
Div. 18. Room 4624.

AMENDMENT

(3

PATENT OFFICE
APR 30 1935
DIVISION 18

Hon. Commissioner of Patents,
Washington, D.C.

S I R : -

In response to the office action of October 30, 1934 in the above entitled application I hereby amend the same as follows:

Rewrite claim 5 as follows:

in combo.

6. In a motor compressor unit, a V-shaped cylinder block, ~~provided with~~ formed upwardly divergent cylinders, inner and outer end heads fitted to said cylinders at opposite ends thereof, slide valve chests formed in the block at the inner sides of the cylinders, slide valves working in the chests for controlling the admission and exhaust of fluid to and from the cylinders and having ports for the delivery of fluid under pressure from the admission points to a storage tank, a substantially V-shaped crank case fitted to the block beneath the cylinders, a counterbalanced crank shaft journaled in the crank case, pistons operating in the cylinders and having rods extended into the crank case, crosshead guides fitted to the sides of said case interiorly thereof, crossheads connecting the rods with the guides and slidable on the [illegible], [illegible] rods operated by [illegible]

(3'

Fig 3 sheet 1

and a system of piping having communication with the chests and the storage tank and having heating means close to the said cylinders.

Fig 2 [illegible]

R E M A R K S .

Applicant has complied with the requirements in the last office action in this case by redrafting the claim to have the same cover the compressor unit. This new claim 6 is directed to the structure disclosed in Figure 2 of the drawings and covers the specific features shown therein. It goes without saying that the references are lacking the specific construction now claimed in this case and this will be apparent on comparing the showing in these references with the disclosure in said Figure 2.

Attention will be called to the construction of the cylinder block and the mounting of the guides and crossheads with the rods connecting the pistons operating in the cylinders with the crank shaft.

In view of this new claim it is thought that the application should now be allowed and an action to this end is respectfully requested at an early date.

Respectfully submitted,

Paul O. Parker

Attorney of record.

Washington, D.C.

April 29, 1935.

Address only
"The Commissioner of Patents,
Washington, D. C."
and not any official by name

DEPARTMENT OF COMMERCE
UNITED STATES PATENT OFFICE
WASHINGTON

All communications respecting this
application should give the serial number,
date of filing, and name of
the applicant

May 10, 1935.

[illegible]ease find below a communication from the EXAMINER in charge of this application.

Conway P. Coe
Commissioner of Patents.

Frank O. Parker
202 Barrister Building
Washington, D.C.

Applicant: B. Neal
Ser. No. 705 964
Filed Jan. 9, 1934
For -Engine

MAILED

Responsive to amendment of April 29, 1935.

Claim 6 now presented is again not free from a confusion of features of both of the structures disclosed in Figs. 2 and 3, one of which discloses a compressor and the other a motor, the two being of separate, distinct and independent status in the art and Office classification. The structure shown in Fig. 2 does not have "slide valve chests" --- and "slide valves" working therein. No combination of the elements enumerated is stated.

The claim is rejected as failing to define the invention with the particularity and distinctness required by Sec. 4888 R. S. to support a patent.

R. E.
Examiner.

MAIL DIVISION
NOV 7 1935
U.S. PATENT OFFICE

10251

DIVISION 9 PAPER NO. 8
NOV 8 1935
U S P

IN THE UNITED STATES PATENT OFFICE

In re: application of
B. Neal,
ENGINES,
Filed January 9, 1934,
Ser. No. 705,964.
Div. 9. Room 4624.

AMENDMENT.

C

Hon. Commissioner of Patents,
Washington, D.C.

SIR: -

In response to the office action of May 10, 1935 I hereby amend the above entitled application as follows:

Rewrite claim 6 as follows:

In a structure of the kind described, a Both V-shaped cylinder block provided with upwardly divergent cylinders, end heads fitted to said cylinders at opposite ends thereof, each head having valved inlets and outlets, a main outlet lead between the cylinders of the block for a storage tank and having lateral branches to the outlets at the inner sides of said heads, one inlet being located at the center of each head at the outer ends of said cylinders while the remaining inlets are at the outer sides of the heads at the inner ends of said cylinders, a substantially Both V-shaped crank case fitted to the block beneath the cylinders, a counterbalanced crank shaft journaled in the crank case, pistons operating in the cylinders and having rods extended into the crank case, crosshead guides fitted to the sides of said case interiorly thereof, crossheads connecting the rods with the guides and slidable on the same and connecting rods operated by the crank shaft and pivoted to the crossheads for reciprocation of the pistons.

C'

10252

Applicant has substituted a new claim for the one previously presented and it is believed that this claim herein is relieved from the objections noted in the last office action and defines the invention with particularity and distinctness required by Sec. 4888 R. S.

Action on the merits of this claim is respectfully requested and an allowance of the application is asked at an early date.

Respectfully submitted,

Paul O. Parker

Attorney of record.

Washington, D.C.

November 7, 1935.

ADDRESS ONLY
THE COMMISSIONER OF PATENTS
WASHINGTON, D. C.

J81

Serial No. 705 964

Ms-9

DEPARTMENT OF COMMERCE
UNITED STATES PATENT OFFICE
WASHINGTON

☞ IN REMITTING THE FINAL FEE GIVE THE SERIAL NUMBER AT THE HEAD OF THIS NOTICE.

Bob Neal December 17th, 1935.

Your APPLICATION for a patent for an IMPROVEMENT in
Compressor Unit
filed Jan.9,1934 has been examined and ALLOWED with 1 claims.

The final fee, THIRTY DOLLARS, WITH $1 ADDITIONAL FOR EACH CLAIM ALLOWED IN EXCESS OF 20, must be paid not later than SIX MONTHS from the date of this present notice of allowance. If the final fee be not paid within that period, the patent will be withheld, but the application may be renewed within one year after the date of the original notice with a renewal fee of $30 and $1 additional for each claim in excess of 20.

The office delivers patents upon the day of their date, on which date their term begins to run. The preparation of the patent for final signing and sealing will require about four weeks, and such work will not be begun until after payment of the necessary final fee.

When the final fee is paid, there should also be sent, DISTINCTLY AND PLAINLY WRITTEN, the name of the INVENTOR, TITLE OF THE INVENTION, AND SERIAL NUMBER AS ABOVE GIVEN, DATE OF ALLOWANCE (which is the date of this circular), DATE OF FILING, and, if assigned, the NAMES OF THE ASSIGNEES.

If it is desired to have the patent issue to an ASSIGNEE OR ASSIGNEES, an assignment containing a REQUEST to that effect, together with the FEE for recording the same, must be filed in this office on or before the date of payment of the final fee.

After issue of the patent, uncertified copies of the drawings and specifications may be purchased at the price of TEN CENTS EACH. The money should accompany the order. Postage stamps will not be received.

The final fee will NOT be received from other than the applicant, his assignee or attorney, or a party in interest as shown by the records of the Patent Office.

NOTICE.—WHEN THE NUMBER OF CLAIMS ALLOWED IS IN EXCESS OF 20, NO SUM LESS THAN $30 PLUS $1 ADDITIONAL FOR EACH CLAIM IN EXCESS OF TWENTY CAN BE ACCEPTED AS THE FINAL FEE.

☞ UNCERTIFIED CHECKS WILL NOT BE ACCEPTED.

Respectfully,

Conway P. Coe

Commissioner of Patents.

Frank O. Parker
202 Barrister Bld'g.,
Washington, D.C.

611—8064 676

25

FMS:MS.

Div. 9 **Room** 4624

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DEPARTMENT OF COMMERCE
UNITED STATES PATENT OFFICE
WASHINGTON

Paper No. 9

All communications respecting this application should give the serial number, date of filing, and name of the applicant

Please find below a communication from the EXAMINER in charge of this application

Commissioner of Patents.

Dec. ,1935.

Applicant Bob Neal

Ser. No. 705 964

Filed Jan.9,1934

For -Engines-

Frank O. Parker
202 Barrister Bld'g.,
Washington, D.C.

DEC 17 1935

In accordance with the provisions of Order No. 2308, dated March 12, 1917, which reads in part as follows:

* * * * * * * *

> Obvious informalities in the application may be corrected by the examiner, but said correction must be in the form of an amendment, approved by the Principal Examiner in writing, placed in the file, and made a part of the record. The changes specified in the amendment will be entered by the clerk in the regular way.

* * * * * * * *

the changes, hereinafter specified, are made by the examiner in the application above identified.

Should these changes not be satisfactory to the applicant, appropriate amendment may be proposed under the provisions of Rule 78, provided the specification has not been printed.

The application has been amended as follows:

Change title to Compressor Unit.

Page 2, line 1, change "an engine" to a compressor.

Page 2, line 5, change "an engine" to a compressor.

R. E. Adams

Examiner.

JAN 16 36 126400 B - Check — 30.00

DIVISION
JAN 16 1936
U. S. PATENT OFFICE

FINAL FEE PAID TO THE COMMISSIONER OF PATENTS

(Be careful to give correct Serial No.)

Serial No. 705,964

INVENTOR: Bob Neal

PATENT TO BE ISSUED TO Bob Neal

NAME OF INVENTION, AS ALLOWED: Compressor Unit

DATE OF PAYMENT: Jan 16th 1936

FEE: Final

DATE OF FILING: Jan. 9-1934

DATE OF CIRCULAR OF ALLOWANCE: Dec. 17-1935

The Commissioner of Patents will please apply the accompanyi[redacted]ndicated above.

Frank O. Parker
Attorney.

SEND PATENT TO
Frank O. Parker
202 Barrister Bldg.
Wash. D. C.

Final fees will not be received from other than the applicant, his assignee or attorney, or a party in interest as shown by the records of the Patent Office, NOR WILL THEY BE APPLIED IN PENDING APPLICATIONS.

11—8678

U. S. GOVERNMENT PRINTING OFFICE: [illegible]

Feb. 11, 1936. B. NEAL **2,030,759**

COMPRESSOR UNIT

Filed Jan. 9, 1934 3 Sheets-Sheet 3

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Fig. 3.

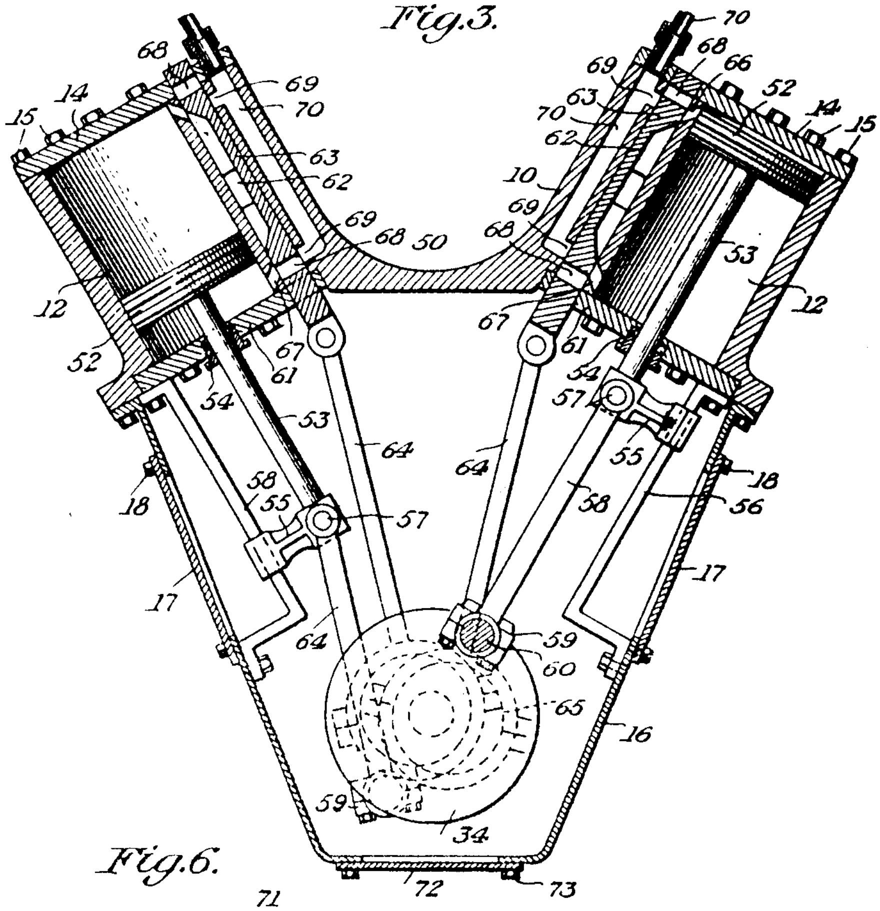
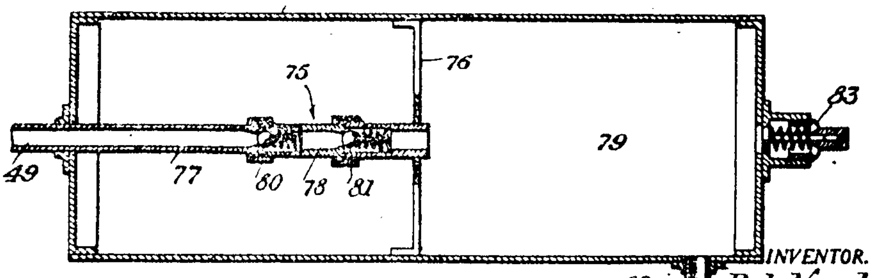


Fig. 6.



INVENTOR.
Bob Neal

BY

ATTORNEY.

Feb. 11, 1936. B. NEAL **2,030,759**

COMPRESSOR UNIT

Filed Jan. 9, 1934 3 Sheets-Sheet 2

Fig. 2.

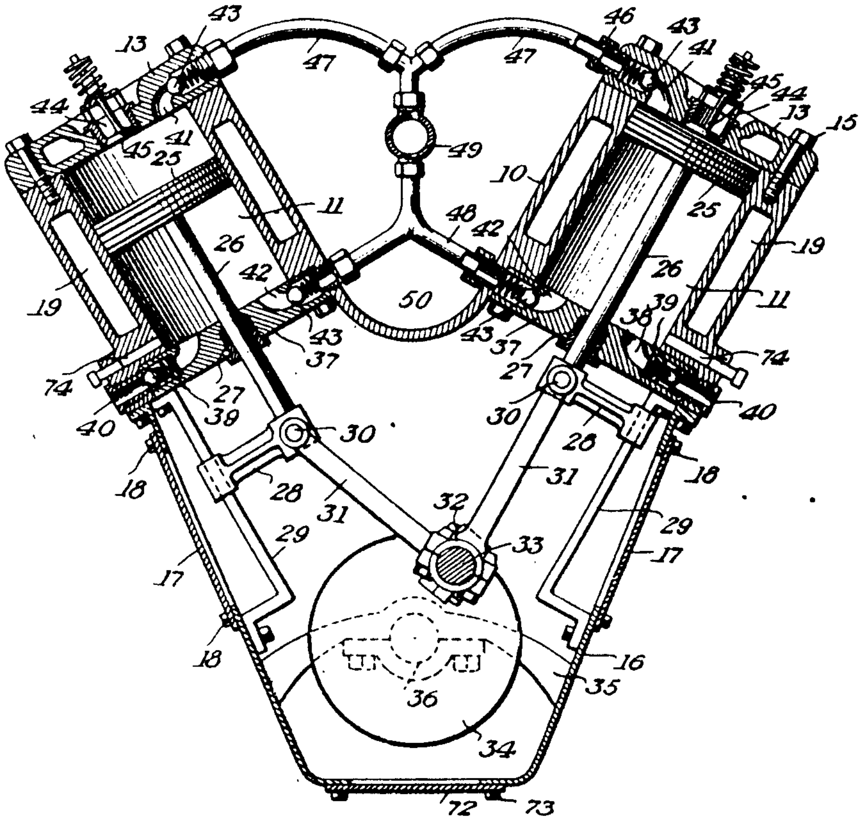
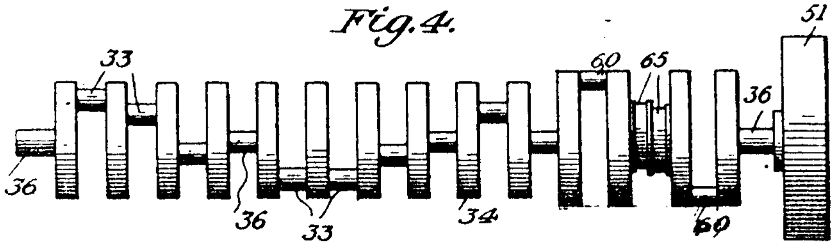


Fig. 4.



INVENTOR.

Bob Neal

BY

ATTORNEY.

Feb. 11, 1936. B. NEAL **2,030,759**

COMPRESSOR UNIT

Filed Jan. 9, 1934 3 Sheets-Sheet 1

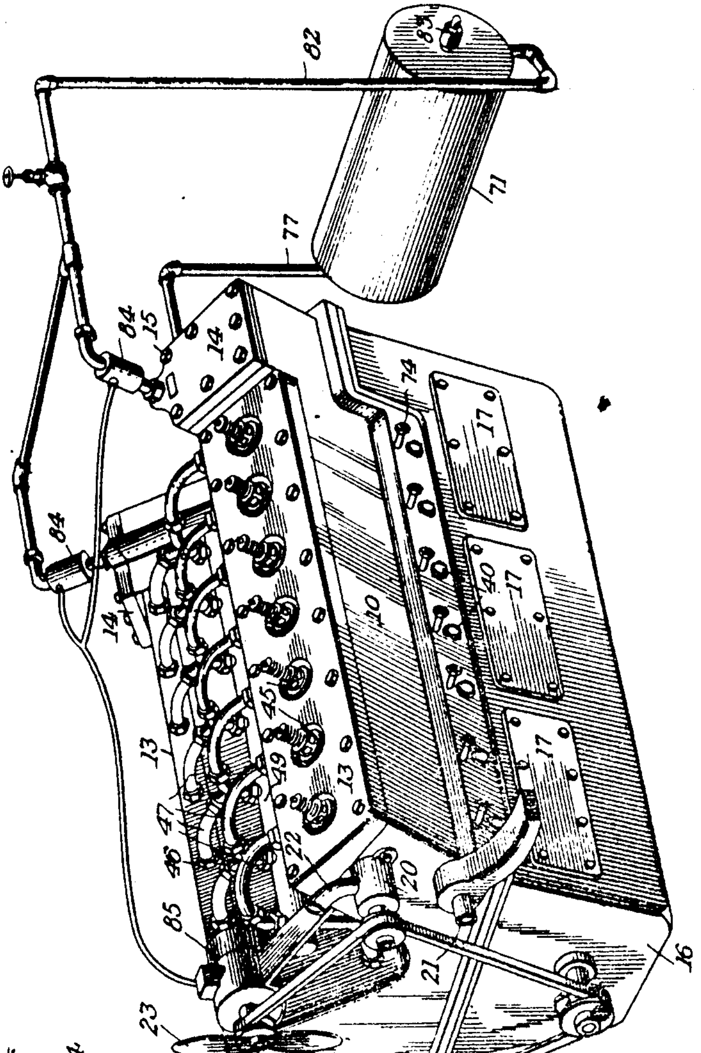


Fig. 1.

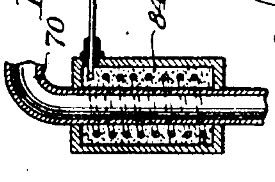


Fig. 5.

INVENTOR.

Bob Neal

BY

ATTORNEY.

Patented Feb. 11, 1936 **2,030,759**

UNITED STATES PATENT OFFICE

2,030,759

COMPRESSOR UNIT

Bob Neal, Arkadelphia, Ark.

Application January 9, 1934, Serial No. 705,964

1 Claim. (Cl. 230—187)

The invention relates to a compressor construction, and more particularly to a combination fluid operated engine and compressor.

The primary object of the invention is the provision of a compressor of this character, wherein there is arranged an automatically counter balanced crank shaft and fluid equalizers within a storage tank, which makes it possible for the said engine to operate on constant reserve tank pressure so as to actuate additional equipment, the pistons for the engine being also automatically balanced and suspended when the said engine is in operation.

Another object of the invention is the provision of an engine of this character, wherein the same is operated from air under pressure, the said air being supplied by compressors, these being in bank with the engine construction.

A further object of the invention is the provision of an engine of this character, wherein the same is of novel construction, as the engine proper and the compressors are operated from the same crank shaft which is of the automatically balanced type, so that high efficiency is attained.

A still further object of the invention is the provision of an engine of this character, which is comparatively simple in construction, thoroughly reliable and efficient in its operation, strong, durable, and inexpensive to manufacture.

With these and other objects in view the invention consists in the features of construction, combination and arrangement of parts as will be hereinafter more fully described, illustrated in the accompanying drawings, which disclose the preferred embodiment of the invention, and pointed out in the claim hereunto appended.

In the accompanying drawings:

Figure 1 is a perspective view of the engine constructed in accordance with the invention.

Figure 2 is a vertical transverse sectional view through the compressor part of the engine.

Figure 3 is a vertical sectional view through the power part of the engine.

Figure 4 is a detail elevation of the crank shaft of the engine.

Figure 5 is an enlarged sectional view through one of the electric heaters for the engine.

Figure 6 is a vertical longitudinal sectional view through the air storage tank including the equalizer.

Similar reference characters indicate corresponding parts throughout the several views in the drawings.

Referring to the drawings in detail the engine in its entirety comprises a cylinder block 10 having formed therein the series of compressor cylinders **11** and the power cylinders **12**, respectively, the block **10** being of the V-type and closing the upper ends of said cylinders are the removable heads **13** and **14**, respectively, which are secured in place by head bolts **15**, as is conventional. Beneath the block **10** is the crank case **16**, which at opposite sides carries the detachable plates **17**, these being held in place by fasteners **18** and such plates are seated so as to be leak proof. The block **10** is chambered to provide a water jacket **19** about the cylinders, while at the forward end of the said block are water pumps **20** circulating water through the inlet pipe **21** which leads into the jacket and letting said water out therefrom through the outlet pipe **22** leading from said water jacket. Next to the pumps **20** is a fan **23** operated from a belt **24** which also drives the pumps.

Working within the cylinders **11** are the reciprocating pistons **25**, their rods **26** being slidable through packing glands **27** and fixed to crossheads **28**, which are slidably mounted upon guides **29** secured within the crank case **16** to opposite side walls thereof. These crossheads **28** are fitted with wrist pins **30** pivotally connecting therewith the connecting rods **31** which by the bearings **32** are engaged with their cranks **33** of a counter balanced crank shaft **34**, which is mounted in supports **35** arranged in the crank case **16**, the shaft being supplied with the required bearings **36**.

The inner ends of the cylinders **11** are fitted with inner end heads **37**, which are provided with air intake ports **38**, these being fitted with spring ball inlet checks **39**, the air having admission through passages **40** opening exteriorly of the block **10**. The glands **27** are associated with the heads **37**.

The heads **13** and **37** are provided with the compressed air outlets **41** and **42**, respectively, these being fitted with spring ball checks **43**, the heads **13** being also provided with the central air inlets **44**, which are fitted with spring checks **45**. By couplings **46** are attached to the air outlets **41** and **42** the outlet feed pipes **47** and **48**, respectively, these leading to a main conduit **49** located in the center channel **50** in said block **10**.

At the rear end of the block **10** and on the shaft **34** is the fly wheel **51**, this being of conventional type.

Working within the cylinders **12** are pistons **52**, their rods **53** sliding through packing glands **54** and fixed in crossheads **55** slidably mounted upon guides **56** which are secured within the crank case 16 at opposite side walls thereof. The crossheads 55 carry wrist pins 57 connecting therewith connecting rods 58, these being engaged by bearings 59 with their respective cranks 60 of the crank shaft 34, the inner ends of the cylinders 12 being also closed by inner heads 61 with which are associated the glands 54.

On the cylinders 12 are slide valve chests 62 in which are the slide valves 63, these being operated by throw rods 64 actuated by cams 65 and such valves controlling the air admission and exhaust of air to and from the cylinders 12 through the ports 66 and 67, and these valves 63 are provided with the ports 68 for the delivery of air under pressure from the inlet passages 69 common to a lead 70 from a compressed air storage tank 71.

The bottom of the crank case 16 is fitted with a removable plate 72 which is secured in place by fasteners 73, and when this plate is removed access can be had to the crank shaft 34 and the bearings for the engine, as well as other parts within said crank case, as should be obvious.

Leading into the cylinders 11 are the passages 74 of a lubricating system (not shown).

The storage tank 71 for the compressed air includes therein a double check discharge nozzle 75, this being supported by a member 76 and leading to this equalizer is an air inlet pipe 77 which has the communication 78 with the chamber 79 formed by said tank. In the equalizer 75 are the spaced spring ball checks 80 and 81, respectively, one being for the inlet side and the other for the exhaust or outlet side of said equalizer. This pipe 77 is connected with the main conduit 49, while a pipe 82 is connected with the leads 70, the tank being also fitted with an automatic relief valve 83 of any approved type.

About the pipes 70 for the passages 69 are the electric heating units 84 which are for the purpose of heating the air under pressure above a freezing temperature when delivered from the tank 71 to the cylinders 12.

Supported on the block 10 is an electric generator 85 which is driven from the shaft 34 through a belt 24 and this generator is included in an electric circuit which also has the heaters 84 so that these will operate from current furnished by said generator.

The storage tank 71 with the equalizer is so constructed that it is possible to pump air into the said tank with a tank pressure of two hundred pounds, while the compressors are only pumping against fifteen pounds or atmospheric pressure. Outside air pressure source can be coupled with the tank to augment that pressure derived from the cylinders 11 of the engine.

What is claimed is:

In a structure of the kind described, a V-shaped cylinder block provided with upwardly divergent cylinders, end heads fitted to said cylinders at opposite ends thereof, each head having valved inlets and outlets, a main outlet lead between the cylinders of the block for a storage tank and having lateral branches to the outlets at the inner sides of said heads, one inlet being located at the center of each head at the outer ends of said cylinders while the remaining inlets are at the outer sides of the heads at the inner ends of said cylinders, a substantially V-shaped crank case fitted to the block beneath the cylinders, a counterbalanced crank shaft journaled in the crank case, pistons operating in the cylinders and having rods extended into the crank case, crosshead guides fitted to the sides of said case interiorly thereof, crossheads connecting the rods with the guides and slidable on the same and connecting rods operated by the crank shaft and pivoted to the crossheads for reciprocation of the pistons.

BOB NEAL.

32

REFERENCE REQUEST—FEDERAL RECORDS CENTERS

NOTE: Use a separate form for each request.

SECTION I—TO BE COMPLETED BY REQUESTING AGENCY

ACCESSION NO.	AGENCY BOX NUMBER	RECORDS CENTER LOCATION NUMBER
612864	OF	15/46-53-3-1 (2/0c)

DESCRIPTION OF RECORD(S) OR INFORMATION REQUESTED

☐ BOX

☑ FOLDER (include file number and title) 2030759

REMARKS

NATURE OF SERVICE

☐ FURNISH COPY OF RECORD(S) ONLY ☐ PERMANENT WITHDRAWAL ☑ TEMPORARY LOAN OF RECORD(S) ☐ REVIEW ☐ OTHER *(Specify)*

SECTION II—FOR USE BY RECORDS CENTER

☐ RECORDS NOT IN CENTER CUSTODY ☐ RECORDS DESTROYED

☐ WRONG ACCESSION NUMBER—PLEASE RECHECK

☐ WRONG BOX NUMBER—PLEASE RECHECK

☐ WRONG CENTER LOCATION—PLEASE RECHECK

☐ ADDITIONAL INFORMATION REQUIRED TO IDENTIFY RECORDS REQUESTED

☐ MISSING *(Neither record(s), information nor charge card found in container(s) specified)*

☐ RECORDS PREVIOUSLY CHARGED OUT TO *(Name, agency and date)*:

REMARKS

DATE	SERVICE	TIME REQUIRED	SEARCHER'S INITIALS
7-10-89			WSS

SECTION III—TO BE COMPLETED BY REQUESTING AGENCY

NAME OF REQUESTER	TELEPHONE NO. ☐ FTS	DATE	RECEIPT OF RECORDS
Gartrell	477	7-6-89	

NAME AND ADDRESS OF AGENCY (Include street address, building, room no. and ZIP Code)

PATENT & TRADEMARK OFFICE
1232 S. EADS STREET
ARLINGTON, VA. 22202

Requester please sign, date and return this form, for the item(s) loaned to you, ONLY if the block to right has been checked by the Records Center. ☐

SIGNATURE

DATE

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36 CFR 1228.162

SEARCHED

Class	Sub	Date	Ex'r
60	59 (U.S.)	Mar. 8, 1934	J.A.K.
230	56 58 157 192	5-10-35	F.M.A.
103	161		

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INDEX OF CLAIMS

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CONT [redacted]

1. Application ✓ papers.
2. Prints Jan. 13/34.
3. REJECTED MAR 12 1934
4. Amdt A Aug. 31 1934
5. Rej. Oct. 30. 1934
6. Amdt B april 24/1935
7. Rej May 10. 1935
8. Amdt C Nov. 7, 1935
9. Extra Amdt D Dec 17, 1935
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